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LIGHT WEIGHT MODULAR MULTI-LAYER INSULATION

by G. E. Nies

UNION CARBIDE LINDE DIVISION

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA LEWIS RESEARCH CENTER

(CONTRACT NAS 3-12045)

James R. Faddoul, Project Manager

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College Park, Md. 20740

NASA CR- 72856

FINAL REPORT

LIGHTWEIGHT MODULAR
MULTI-LAYER INSULATION

By

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UNION CARBIDE CORPORATION
LINDE DIVISION
Tonawanda, New York 14150

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

February 26, 1971

CONTRACT NAS 3-12045

NASA Lewis Research Center
Cleveland, Ohio
James R. Faddoul
Project Manager
Chemical Rockets Division Office

ABSTRACT

The design of a Self Evacuating Multilayer Insulation (SEMI) system for a 10 ft. (3.05M) diameter by 20 ft. (6.1M) long liquid hydrogen tank is described. The design, fabrication and testing of a model system (30 inch (.76M) in diameter) simulating the full size insulation system is also presented. Model system testing included both thermal and structural evaluations. Thermal tests, using liquid hydrogen, were performed at NASA Plumbrook Station, "K" site Sandusky, Ohio while the structural tests were performed at NASA Goddard Space Flight Center, in the Launch Phase Simulator Facility, Greenbelt, Maryland. The SEMI system was subjected to a combined launch vacuum profile to simulate a typical launch profile. The system performed as designed. The SEMI system materials, consisting of composite layers of rigid open cell polyurethane foam and double aluminized Mylar radiation shields were unaffected by the dynamic testing. A heat flux of 1.3 Btu per hour (4.1 Watts per M^2) was measured before and after the dynamic tests. The three panel, 18 radiation shield system offers a lightweight (0.39 lb./ft^2, 1.9 Kg/M^2) cryogenic insulation.

The work was performed under contract NAS 3-12045 and reported in CR-72856.

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1.0 SUMMARY

The objectives of this program were (1) to design a full scale lightweight self evacuating multilayer insulation (SEMI) system for liquid hydrogen tankage considering thermal and structural loads, and (2) to design and fabricate a scale model of the full scale system for thermal and structural evaluation tests

A SEMI system was designed for a so called "full size" tank, namely a tank 10 feet (3.05 M) in diameter by 20 feet (6.1 M) in length with hemispheric heads. For evaluation, the tank also included an inner skirt support interstage structure) insulated with polar panels. The structural and thermal loadings imposed on the insulation system were determined during the design phase.

A scaled down version of the full size insulation system referred to as the "model system" was designed and fabricated by UCC Linde Division. The insulation system consisted of carbon dioxide (CO_2) filled panels composed of seven composite layers of open cell rigid polyurethane foam spacers and six aluminized Mylar radiation shields enclosed in a 4-ply aluminized Mylar composite casing, protected against air exposure by use of an impermeable Mylar aluminum foil composite at all external air exposure surfaces. The scale model insulation system was of a three layer design with panels circumferentially wrapped on the cylindrical portion. The forward bulkhead and simulated skirt section was insulated with panels installed in a polar shingle fashion. The panels were attached to each other and to the tank using VELCRO fasteners. Panel to panel seals were completed with a foil tape, and edge sealed with silicone rubber. The tank used for this effort as well as the handling cart was designed, fabricated and proof tested by UCC Linde Division. Vibration testing of the bare tank was performed by Dayton T. Brown Test Laboratory.

Subscale testing performed to evaluate system components included vibration testing of 12 inch (.305 M) by 18 inch (.45 M) SEMI panels; quality control checks on casing materials, foam spacers, and radiation shields; and evaluation of VELCRO attachments.

Thermal testing of the model system was performed at NASA Plumbrook Station, Sandusky, Ohio, and dymanic testing was performed at NASA Goddard Space Flight Center, Greenbelt, Md. After dynamic and thermal testing the Model System at NASA facilities, Linde performed the post test evaluation at NASA Lewis Research Center, Cleveland, Ohio.

The system performed as designed with the thermal performance being the same when measured before and after the dynamic tests (dynamic testing was a combined acceleration, vibration, accoustic and launch vacuum profile evaluation). A heat flux of 1.3 Btu per hr. ft.^{2} (4.10 watts per M^2) was measured for both tests in the space condition "insulation recovered" mode, (See Section 4.2.3.3). This performance was slightly better than that predicted by the

computer program using flat plate data, corrected for panel edge effects. During dynamic testing, pressure build-up behind the panel (of an unknown cause) resulted in casing failure on two panels, and minor spacer damage to one of the panels. Except for this damage, the system functioned nominally in both mechanical and thermal performance.

2.0 INTRODUCTION

Storage of cryogenic liquids plays an important role in the further exploration of space. Development and utilization of a light weight high performance insulation is necessary to reduce launch lift off weights, thus permitting extended space missions without suffering excessive weight or thermal penalties. Aware of these requirements, Union Carbide Corporation, Linde Division, under the direction of the NASA's Lewis Research Center at Cleveland, Ohio, has been working on the development of a Self Evacuating Multilayer Insulation (SEMI) system for the past few years. This report covering work performed under contract NAS 3-12045, is a logical extension of the previous work performed by UCC Linde Division on contracts NAS 3-6289 (Ref. 1) and NAS 3-7953 (Ref. 2), in that it provided an evaluation of the SEMI system exposed to a launch environment, including combined vibration, acceleration, accoustics and vacuum profile, (pressure decay during simulated lift off). System evaluation consisted of visual observation for damage as well as thermal performance testing with liquid hydrogen, before and after dynamic testing.

The SEMI system consists of the following components, a vacuum tight casing or bag, open cell rigid urethane foam spacers, highly reflective double aluminized mylar radiation shields and a cryopumpable fill gas (Coleman grade carbon dioxide, CO_2). In order for a cryopumpable insulation to become operable, a portion of each panel must contact the cryogenic surface in order that upon cooldown of the vessel, the gas cryopumps on the cold end of the panel, thus reducing the pressure of the panel to the vapor pressure of the gas commensurate with the temperature of the cold surface. Furthermore, for each panel to contact the cold tank surface, the panels must of necessity be shingled. UCC has investigated cryopumpable shingled three layered system of panels, as illustrated in Figure 1 and 2.

This program was divided into two major areas, namely design of a full scale insulation system for a 10 ft. (3.05 M) diameter by 20 ft. (6.1 M) long tank, and design, fabrication and post test evaluation (after completion of the NASA performed test phase), of a model insulation system 30 inches (.75 M) in diameter.

The full scale insulation system design effort was performed to investigate panel layouts and attachments, in relation to thermal performance, as well as the structural requirements and capabilities of the SEMI system. In support of the analysis, several small scale tests were performed to provide needed test data. A complete writeup of the full scale insulation work is presented in Section 4.1.

A similar investigation, including the design, fabrication, and testing of a model system that simulated the full scale insulation system was performed. The purpose of testing a model system (and chief purpose of this contractural effort) was to determine if any thermal degradation of the SEMI system occurred following exposure to the complete launch environment. A complete write-up on the model system and related hardware efforts is presented in Section 4.2.

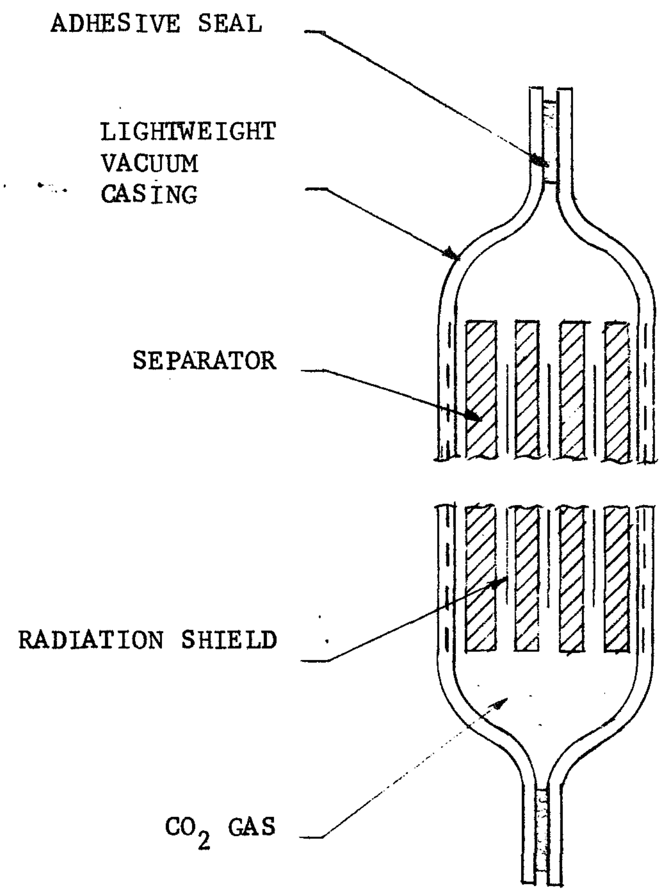


FIGURE 1. SCHEMATIC OF SEMI PANEL SHOWING COMPONENT MATERIALS

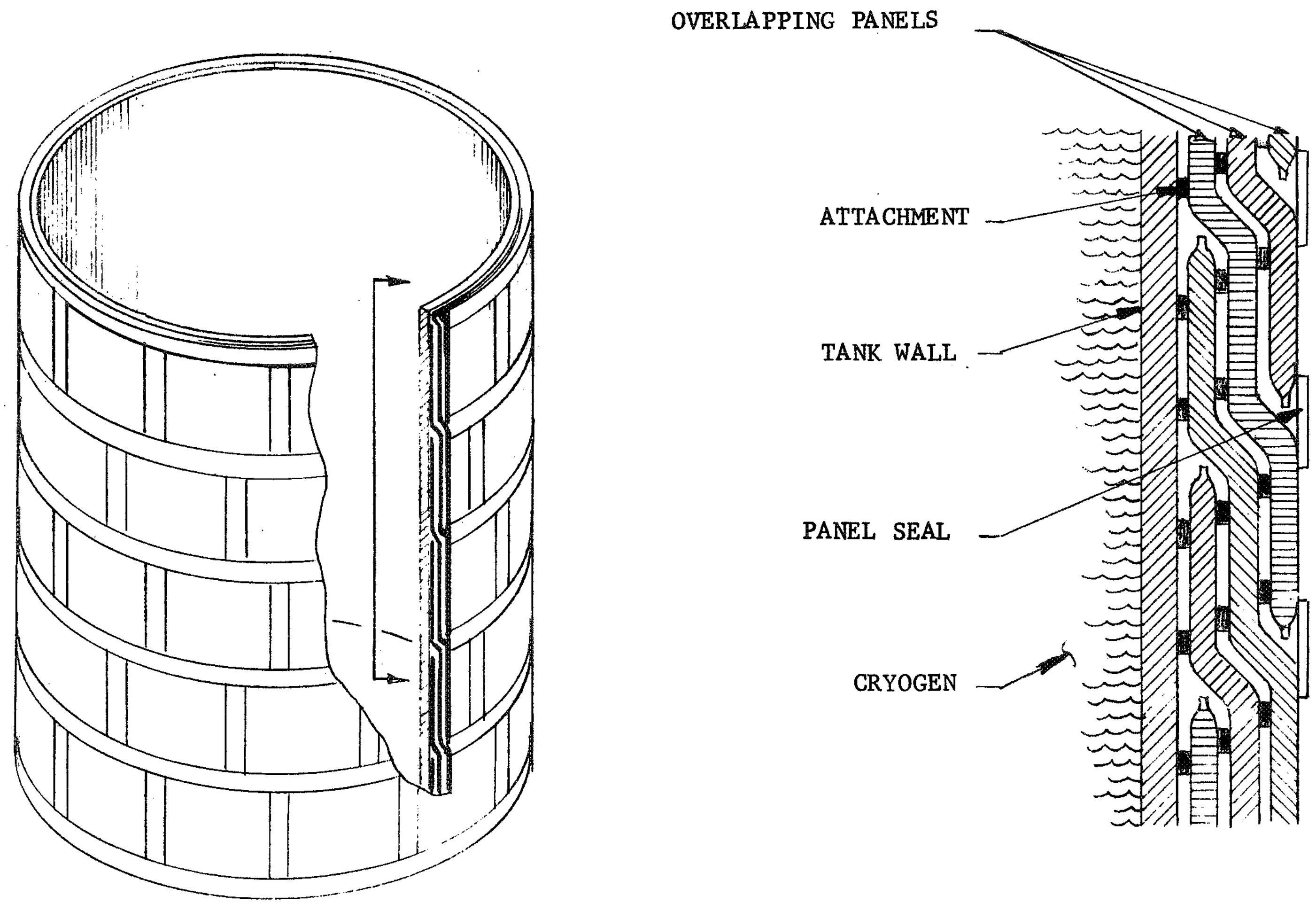


FIGURE 2. SEMI PANEL SHINGLE ARRANGEMENT INSTALLATION

Thermal testing of the Model system was performed at "K" site, Plumbrook Station, Sandusky, Ohio, and dynamic testing was performed in the Launch Phase Simulator, at Goddard Space Flight Center, Greenbelt, Maryland. Testing of the model system at both of the previously mentioned NASA facilities, was performed entirely by NASA personnel.

The work on this contract was performed under the direction of the National Aeronautics and Space Administration Chemical Rockets Division, Lewis Research Center Cleveland, Ohio. The technical monitor was Mr. James R. Faddoul. The UCC Linde Division program manager was Mr. G.E. Nies.

Other Linde Division personnel contributing to this effort included Mr. L.R. Niendorf, Mr. L.K. Eigenbrod and Mr. E.S. Kordyban (structural analysis), and Mr. L.D. Potts and Mr. D. Suckow (thermal analysis).

3.0 DISCUSSION OF RESULTS

The design of a Self Evacuating Multi Layer Insulation (SEMI) system for a large liquid hydrogen (LH_2) tank was completed. As a result of this study, it can be stated that a SEMI system consisting of reasonably sized panels could be installed on LH_2 tankage, and that furthermore the overall performance of SEMI panels is not extremely sensitive to the ratio of the edge perimeter/panel area for large panels. A performance of 0.6 Btu/hr.-ft.^{2} (1.9 watts/M^2) and 7.45 Btu/hr.-ft^2 (23.47 watts/M^2) for the recovered and compressed insulation condition respectively was determined to be achievable for this size of tankage, 10 ft. (3.05 M) diameter x 20 ft. (6.1 M) overall length).

A ratio of 4 square inches (.002 M^2) of VELCRO closure per ft.^{2} (.09 M^2) of SEMI panel was determined to provide acceptable casing stress levels for the expected 8.5 g combined longitudinal launch loadings.

Thermal and dynamic testing of the SEMI insulated Model Tank, designed to simulate the full size SEMI system was completed. Testing of the Model System indicated nominal thermal performance for the insulation prior to and after the dynamic environment. An un-explainable pressure build-up behind the panels, (between the panels and the tank) resulted in casing failure on two of the eight circumferential panels, including minor spacer damage to one of the two failed panels. The remaining six circumferential panels did not appear to have suffered any damage during the combined simulated launch test program. During disassembly, it was noted that the circumferential panels, with the exception of the panels damaged by the over pressure, were still intact. The three polar panels were failed, however that was also attributed to overpressure behind the panels. The foam spacers and shields suffered no damage during tests. A heat flux of 1.3 Btu/Hr.-Ft.^{2} (4.1 watts/M^2) was measured in thermal tests of the recovered insulation system, i.e. space condition, before and after the dynamic testing.

Sealing between panels, and maintaining the space behind the panels was again a troublesome area. With the present techniques, this operation is time consuming and quite prone to failure. Failures can occur as a result of folds in the casing materials, cracking of the seal material upon exposure to near cryogenic temperatures, or cracking due to casing flexure upon evacuation. Sealing between panels requires additional development effort in the future.

3.1 Conclusions

The SEMI system of cryopumpable pre-fabricated panel insulation is definitely applicable to hydrogen tankage. This was demonstrated thermally in contracts NAS 3-6289 and NAS 3-7953, and again under this contract, where it was also demonstrated that the thermal performance and the SEMI system, is unaffected by the rigors of launch.

Using VELCRO closures to attach the panels to each other and the tank has been demonstrated to be an acceptable technique. However, the resistance of the 4 ply Mylar casing material to delaminate under the peel stress from the VELCRO should be investigated and improved if possible. Several delaminations occurred at disassembly, (Post Test Evaluation) although there was no evidence of delamination during dynamic testing.

The insulating materials, namely, double aluminized Mylar radiation shields and the thin sliced rigid open cell polyurethane foam spacers, are acceptable for service in vibration, acceleration and accoustic environments.

One definite problem exists, and that is that a method of satisfactorily sealing between the panels must be developed if the SEMI system is to be successfully applied, especially if re-usable vehicle requirements are to be met.

4.0 ANALYSIS AND TEST PROCEDURES

4.1 Full Scale Insulation System Design

One objective of this program was to design a Self Evacuating Multilayer Insulation system (SEMI) for liquid hydrogen (LH_2) tankage. Specifically, the design was for a LH_2 tank, 10 feet (3.05M) in diameter by 20 feet (6.1M) long, employing an inner stage skirt at one end.

The SEMI system was developed by UCC, Linde Division under contracts NAS3-6289 and NAS3-7953. In the present form, it consists of 6 radiation shields and 7 thermal spacers enclosed within a flexible casing. This bag is filled with Carbon Dioxide gas at atmospheric pressure, which upon exposure to cryogenically cooled surface will condense to a solid state, exhibiting a vapor pressure of less than 0.1 micron (1×10^{-4} torr). (The pressure of pure Carbon Dioxide is 1×10^{-8} torr at liquid nitrogen temperature -320°F. (77°K.)).

The panels are installed in a three layered shingled fashion. This permits each panel to contact the cryogenic surface, thus producing the necessary cryopumping phenomena. This system provides an insulation with minimum heat transfer by radiation, solid conduction and convection for both ground hold and space conditions, but without the vacuum pumping or purge equipment associated with other systems. The following subsections will describe the design work performed in the areas of panel layout, panel thermal analysis and panel structural analysis. More complete write ups on the structural and thermal analysis are included in the appropriate referenced appendices.

4.1.1 Panel Layout

Panel configurations for the Full Scale Insulation System are divided into three major groupings according to tank area covered:

(A) Polar shingled panels. (For use on tank heads).

(B) Support skirt or inner skirt panels. (For use on tank skirt).

(C) Outer or circumferential panels. (For use on tank circumference)

Several panel layout configurations (see Figure 3) have been determined for each of the three areas. In Figure 3 the column heading "Panel System" refers to the number of panels that are required to complete one layer of panels on the tank for that particular section. To complete a 3 layered system, three times as many panels are required. For example under Section A, polar panels, sizes for a 3 panel, 6 panel and 9 panel system are listed. Final choice of a panel system for this section from the three available "Panel Systems" listed in Figure 3 would then be evaluated based upon achieving a design that allows economic and realistic panel sizes in regards to handleability and maximizing the thermal performance. This same explanation holds true for the Inner Skirt

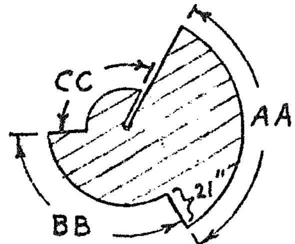
FIGURE 3

PANEL CONFIGURATIONS - FULL SCALE INSULATION

TASKI - NAS 3-12045

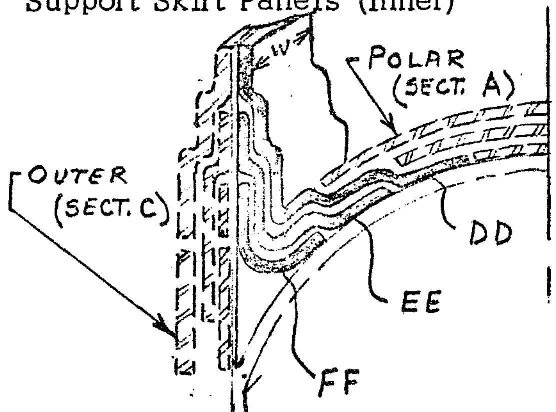
A. Polar Shingled Panels

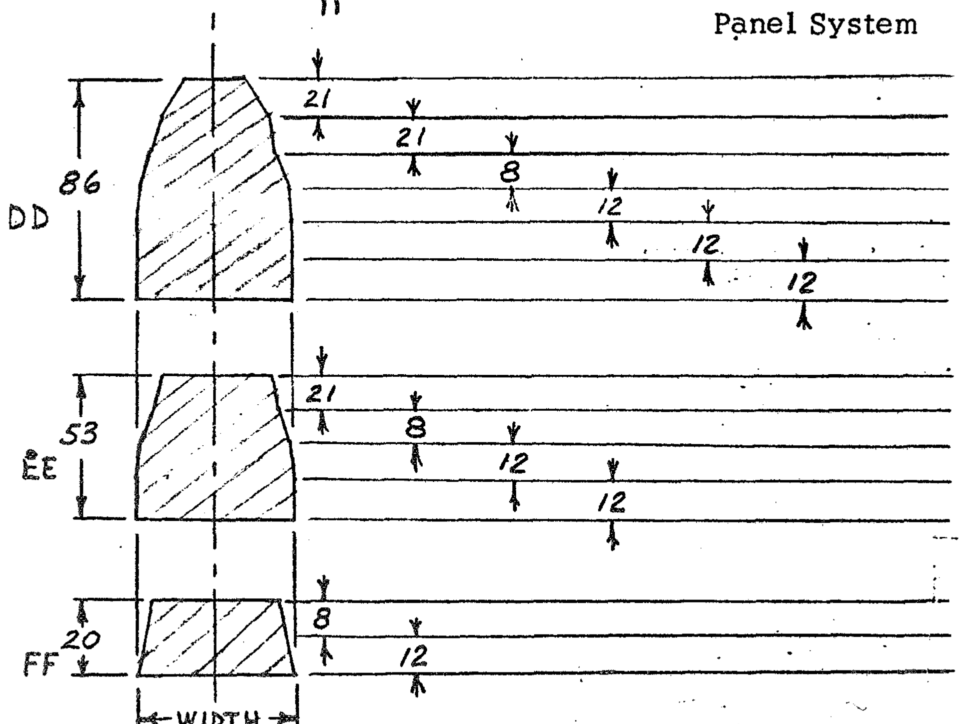
60-inch Spherical Radius



Panel System	Total Panels R'qd	Panel Size (Inches) AA	BB	CC
3	3	109	82	44
6	6	55	41	22
9	9	36	27	15

B. Support Skirt Panels (Inner)

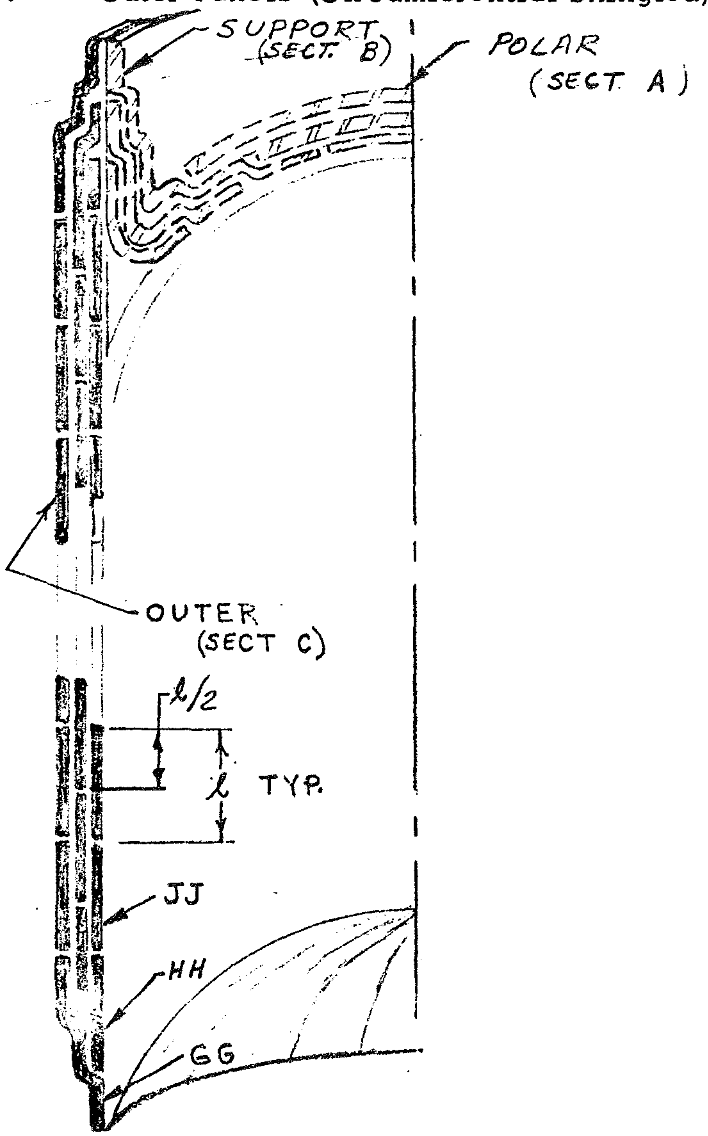


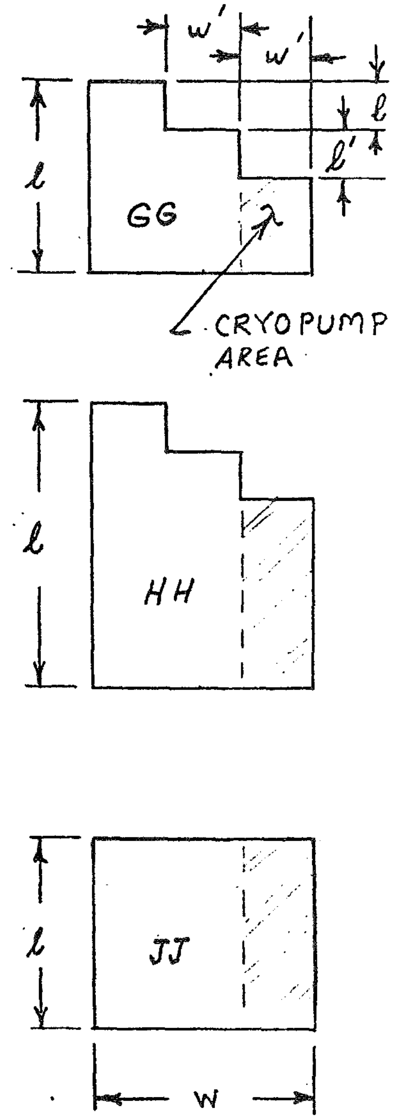


	Panel Width - Inches		
Panel System	3	6	9
DD	44	22	15
	82	41	27
	109	55	36
	119	60	40
	126	63	42
	126	63	42
	126	63	42
EE	82	41	27
	109	55	36
	119	60	40
	126	63	42
	126	63	42
FF	109	55	36
	119	60	40
	126	63	42
Total Panels Required	9	18	27

FIGURE 3 (Cont'd.)

C. Outer Panels (Circumferential Shingled)





Panel System	Style	No. Req'd	Panel Size (Inches) ℓ	w	Step ℓ'	w'
12	GG	12 (6 each end)	60	96	12	32
	HH	12 (6 each end)	90	96	12	32
	JJ	18	60	96	-	-
18	GG	18 (9 each end)	24	63	6	21
	HH	18 (9 each end)	36	63	6	21
	JJ	135	24	63	-	-

panels (Section B) and the Circumferential panels (Section C). This does not imply that only like panel systems from the various sections can be used in the final design. What this does mean is that panels having dimensions similar to that of "A" (A, Figure 3) will require that number of panels wrapped in polar fashion to provide the necessary three layered shingled polar cap insulation. For example, using Section A, a three panel section of polar caps could be combined with a six panel system for the inner support skirt (Section B) and the twelve or eighteen panel system for the outside circumferential panels (Section C). Likewise in Section B, panel style FF in any configuration is only 20 inches (.51M) wide, and therefore, it is quite feasible to use the three panel system for FF (panel size 20 in x 126 in) (.51M x 3.2M) while using the nine panel system for style DD (panel size 42 in x 86 in) (1.07M x 2.18M) in order to permit utilizing larger panel sizes.

In regards to sizing the circumferential panels, Section C, the panel system number must be an even number i.e. 4, 6, 8 etc. in order that the three layered and the stepped end conditions can be accomplished. The panel stepped end is an attempt to match the panel temperature profile to that of the support skirt. In order to maintain a half lapped joint configuration to reduce radiation losses, every other panel (style HH) must be half again as long as its adjacent panel (style GG).

4.1.2 Thermal Analysis

The thermal performance of a shingled insulation system for hydrogen tankage with an interstage support skirt was analyzed. For purposes of the analysis, the tank is regarded as a cylinder, 10 ft. (3.05M) long and 10 ft. (3.05M) in diameter, with hemispherical heads. The overall length is 20 ft. (6.1M).

The analysis of such a system is complicated by the fact that only a portion of the heat influx occurs in the ordinary manner through the panels. Indeed, except near the edges, conductivities of such insulation panels in a direction normal to their radiation shields are generally very low and if all heat influx could be confined to this mechanism, the insulation task (and problem of analysis) would be minimal. Panel edge effects would be strictly local in character and might be rendered insignificant by discrete stacking of the panels to conceal the butt joints. In fact, however, even though the panels are carefully overlapped to minimize edge effects, the casings and radiation shields provide alternative heat paths which, through comparatively long and tortuous, become significant because of the relatively high lateral conductivities involved. Thus, the edge effects are by no means local, and analysis of such a system is a fully three dimensional problem of vast scope, complicated because of its highly implicit character; the thermal flux at any point generally depends on temperatures everywhere else in the system. Mathematically, this means that the analysis requires enormous sets of simultaneous equations. The associated problems of data storage and round off error are nearly inseparable unless we exploit the structure of the system, to "whittle" the problem down to a tractable size.

There is, of course, a close analogy between the actual physical structure of the system and the mathematical structure of the governing equations. Thus, even before writing down any equations, we can effect most of our economics via physical observations. The first thing to be noted is that, because of longitudinal symetry, we need analyze only half of the actual system. Thus, we can imagine a plane normal to the central axis and passing through its midpoint. The intervention of this plane with the insulation layers can be interpreted as an adiabatic boundary.

Next to be observed is that the insulation temperatures will possess some periodicity in a circumferential direction, depending on the manner in which the panels are wrapped. Thus, panels on the cylindrical portion are so overlapped circumferentially that the structure repeats itself every 60° for the 6 panel system. Similar periodicities exist in the hemispherical portions, but here a wider variety of panel sizes can be employed; depending on the panel size, periodicities of 30°, 60°, or 120° may prevail in the structure. It seems needless to analyze more than one basic interval in the circumferential direction, but it should be noted that the cylindrical and spherical portions interact and thus may display temperature periods which are multiples or fractions of their basic physical periodicity.

A possible simplification suggests itself here, however, related to the way in which the cylindrical and spherical portions are physically joined. Indeed, the insulation panels of the cylindrical portion nowhere directly contact those of the hemispherical portions; at their apparent junction they are separated by the fiberglass support skirt. Because it is relatively thick (0.060 inch) (1.5×10^{-3}M) and conductive (0.2 Btu/ft. hr °F. (1.25×10^{3} joule per M^2 sec°K), the skirt should indeed act to attenuate the periodic behavior of temperatures in circumferential direction. Thus, it seems reasonable to regard the fiberglass as an infinitely good conductor in the circumferential (not, of course, the longitudinal) direction. In this way we can dispense with interference of periodicities between temperature patterns in the cylinder and head. Any small error introduced by this assumption regarding conductivity of the fiberglass skirt will be on the conservative side, i.e., it will tend to over estimate heat leak into the system. The skirt assumption is strategic in more ways than one, as will be shown.

Thus in summarizing, a program was prepared, expressing the heat transfer terms of temperatures along the fiberglass support skirt, which is a common and logical interface between the circumferential and the polar shingle panels, it was more simple to write a new program rather than attempt to rewrite the program used in the previous calorimeter analysis from NAS 3-6289. The polar panel program analysis from contract NAS 3-7953 was modified to account for the cylindrical support skirt rather than the conical support structure used on that contract. A brief description of these changes are listed below.

The technique for computing the temperatures within the insulation system on the hemispherical ends and inside surfaces of the support skirt is outlined in Appendix No. 5, NASA CR 72363, which describes an earlier, similar analysis on the 82.6 inch (2.09M) spherical tank. The technique essentially consists of setting up a finite-difference mesh within the insulation and discretizing the steady-state Fourier heat conduction equation at each mesh point, thereby generating a system of linear equations. Since the coefficient matrix for this system of equations is block-tridiagonal, the equations can be solved using a matrix analogue of the Gauss Elimination technique for tridiagonal systems.

The essential differences between the two analyses are described below. In the original analysis, the temperatures within the support structure were unknown and variable in the circumferential direction. In the present analysis, these temperatures are assumed to be independent of circumferential position and to vary linearly between ambient and cryogenic. These temperatures are thus known and serve as boundary conditions for the system of equations. Also, whereas the original analysis includes the insulation on two hemispheres plus both sides of the support skirt, the present analysis only includes one hemisphere and the inside surface of the skirt. (The outside of the skirt is part of the cylindrical analysis.)

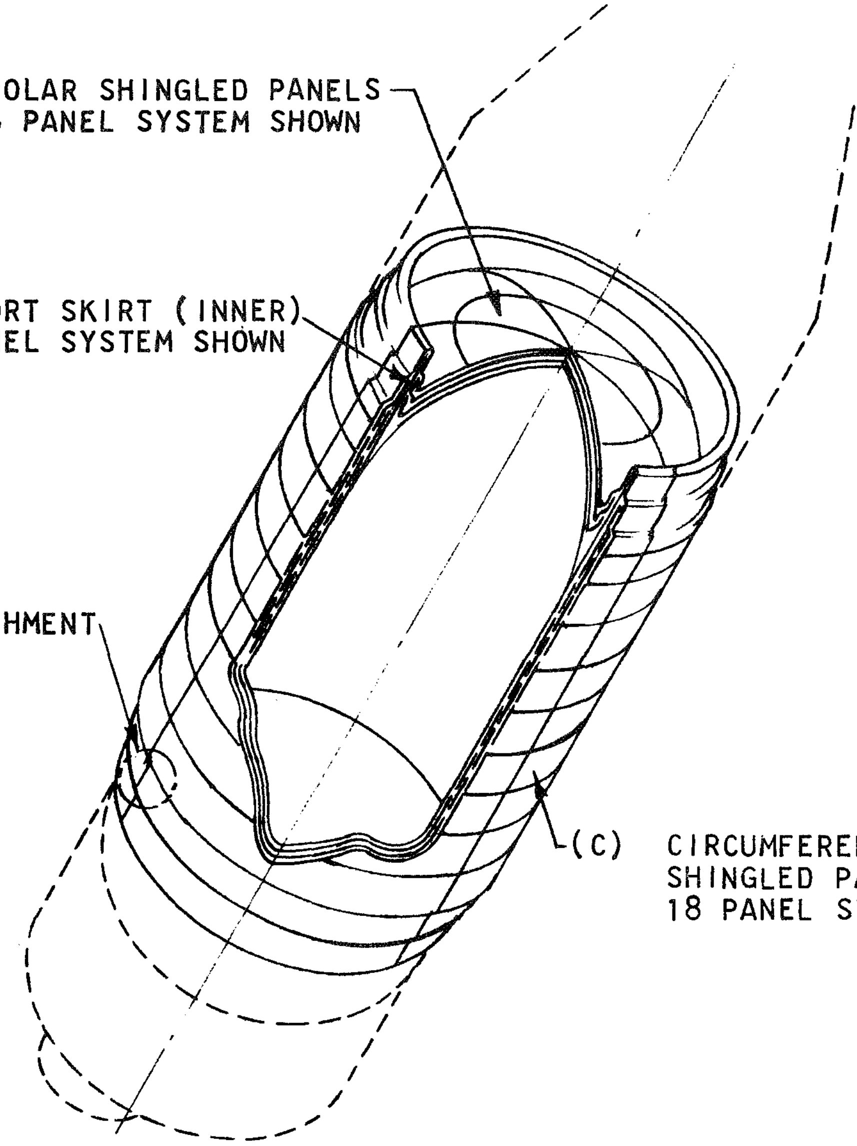
In both analyses, the insulation system divides into "zones" due to the changing overlap configuration. The hemisphere insulation divides into four zones, the skirt insulation into three. However, in the original analysis the number of longitudinal mesh positions was fixed at two per hemisphere zones and one per skirt zone for a total of 11. In the present analysis, the number of positions is variable from 1 to 10 for all zones. Also, the angular distance between circumferential mesh positions has been halved in the present analysis, from 15° to 7-1/2°, thus permitting a finer mesh and consequently more accurate results.

Computer results of three systems of various panel sizes, (Panel sizes are listed in Section 4.1.1) are presented in Table 1. The three cases as presented are the results of the computer analysis indicating the panel combinations leading to performance classified in Table 1 under Computer Analysis as:

a. Best
b. Realistic
c. Poorest

In the "best" performance category, the panel system was designed to minimize joint heat leak, and use the largest possible panels consistent with achieving a cryopumping design. At the other extreme, the panels were selected to achieve small panel sizes, thereby providing maximum handling ease, thus reducing thermal performance. The third case, labeled "realistic" attempted to achieve cryopumpable design, while considering both handling and performance requirements in terms of panel sizes.

Results of the computer analysis for the final design (realistic case) of the full scale insulation shown on Figure 4 (SK-106295) are 4017.2 Btu per hour (1177 Watts) for the compressed case (ground hold) and 323.7 Btu per hour (94.8 Watts) for the uncompressed or space condition. This is determined to be a heat flux of 7.45 Btu per hour ft^2 (23.47 Watts per M^2.) and 0.60 Btu per hour ft.^{2} (1.89 Watts per M^2) respectively. Figure 5 shows the effect the number of head panels has on performance.



DIAMETER - 10 FEET

LENGTH - 20 FEET

FULL SIZE SHINGLED INSULATION SYSTEM FOR HYDROGEN TANKAGE WITH INTERSTAGE SUPPORT SKIRT (TASK I)

CONTRACT NAS3-12045

FIGURE 4
SHEET 1 OF 4

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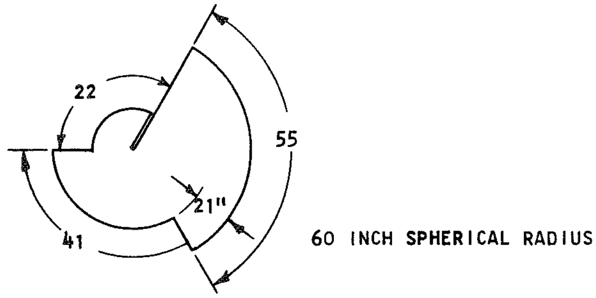
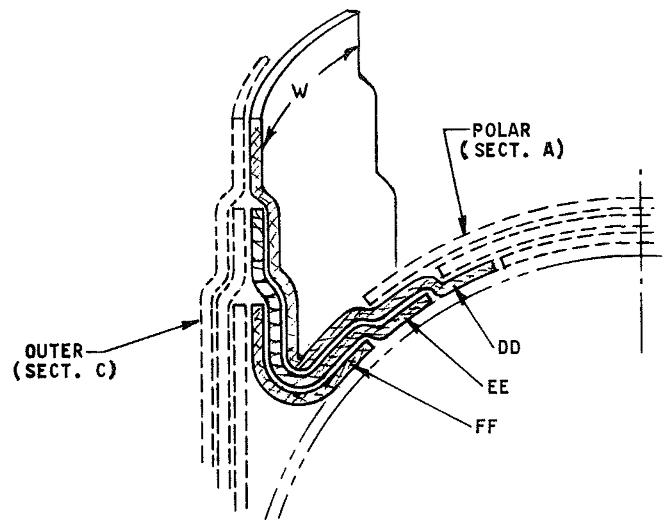
TON[illegible]

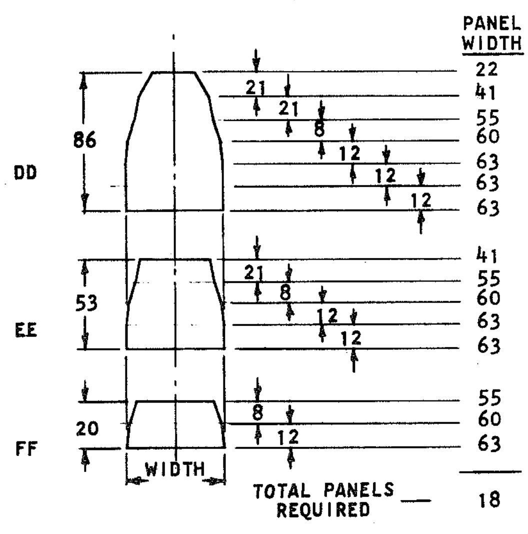
REH | 9-30-69 | |
— | A | SK-106295

(A) POLAR SHINGLED PANELS
6 PANEL SYSTEM SHOWN, 6 REQ'D.



(B) SUPPORT SKIRT PANELS (INNER)





6 PANEL SYSTEM

SHEET 2 OF 4

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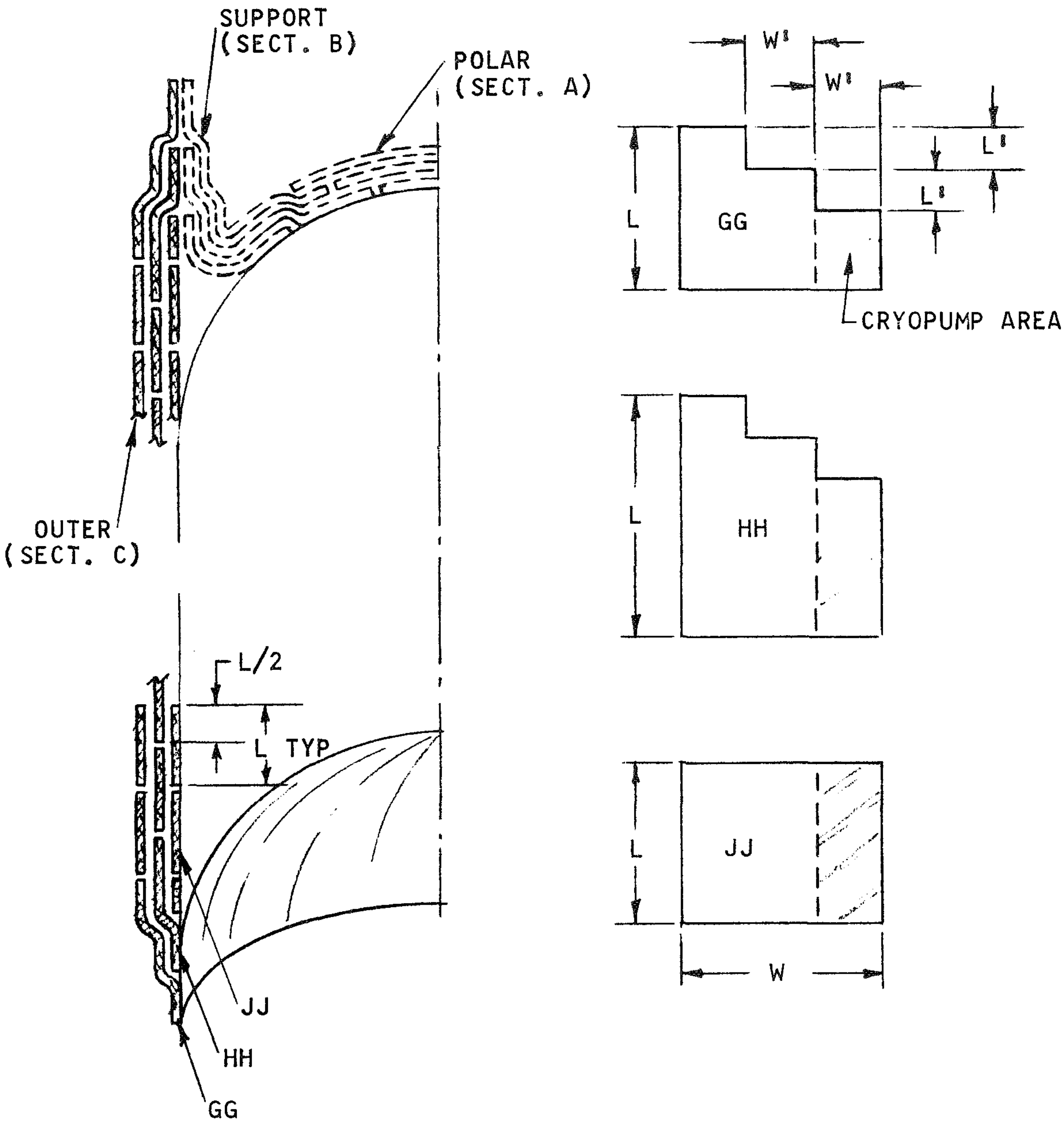
TONAWANDA, N. Y.

DRAWN	DATE	CH'K'D	APP'V'D
REH	9-30-69	SEM	SEM
SCALE —	A	SK-106295	

(C) OUTER PANELS
(CIRCUMFERENTIAL SHINGLED)



PANEL SYSTEM	NO. REQ'D.	PANEL SIZE (INCHES) L	W	STEP L'	W'
18 GG	18 (9 EACH END)	24	63	6	21
HH	18 (9 EACH END)	36	63	6	21
JJ	135	24	63	-	-
	TOTAL REQ'D. 171				

SHEET 3 OF 4

ENGINEERING DEPARTMENT
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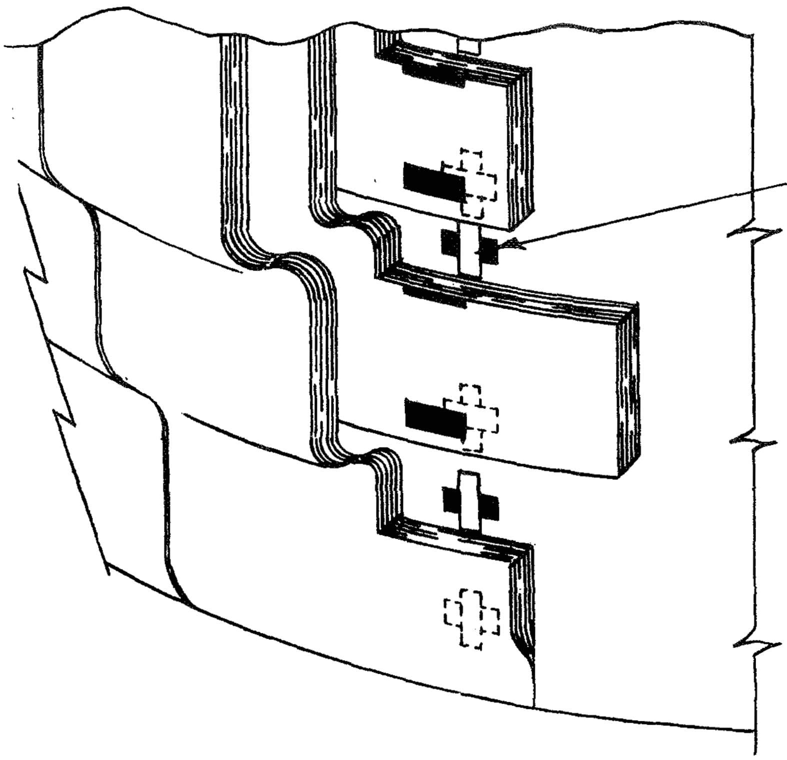
TONAWANDA, N. Y.

REH | 9-30-63 | [illegible] | [illegible]
— | A | SK-106295

(D) ATTACHMENT (VELCRO) TYPICAL



ASSEMBLY NOTES

1. VELCRO TO BE LOCATED ON EACH PANEL IN SUCH A PATTERN THAT EACH SQUARE FOOT OF INSULATION PANEL CONTAINS 4 IN^2 OF VELCRO CLOSURE.

2. DISTANCE FROM CLOSEST EDGE OF VELCRO ATTACHMENT TO EDGE OF PANEL TO BE 2" ± 1/4".

3. SURFACE TO BE WIPED WITH METHYL ETHYL KETONE (MEK) AND ALLOWED TO AIR DRY, PRIOR TO PLACING VELCRO. VELCRO ADHESIVE BACKING TO BE RE-CONSTITUTED WITH A SINGLE MEK WIPE, AND ALLOWED TO BECOME TACKY (APPROXIMATELY 3 MINUTES) BEFORE POSITIONING. ADHESIVE WILL AIR CURE AT AMBIENT TEMPERATURES AND PRESSURE IN 24 HOURS. ALUMINUM SURFACES SHOULD BE MEK WIPED, ABRAIDED WITH A WIRE BRUSH, AND THEN REWIPED WITH MEK AND AIR DRIED PRIOR TO PLACING VELCRO AS PREVIOUSLY DESCRIBED.

SHEET 4 OF 4

ENGINEERING DEPARTMENT LINDE DIVISION OF UNION CARBIDE CORP TONAWANDA, N. Y.			
DRAWN REH	DATE 9-30-69	CH'K'D SCM	APP SCM
SCALE —	A SK-106295		

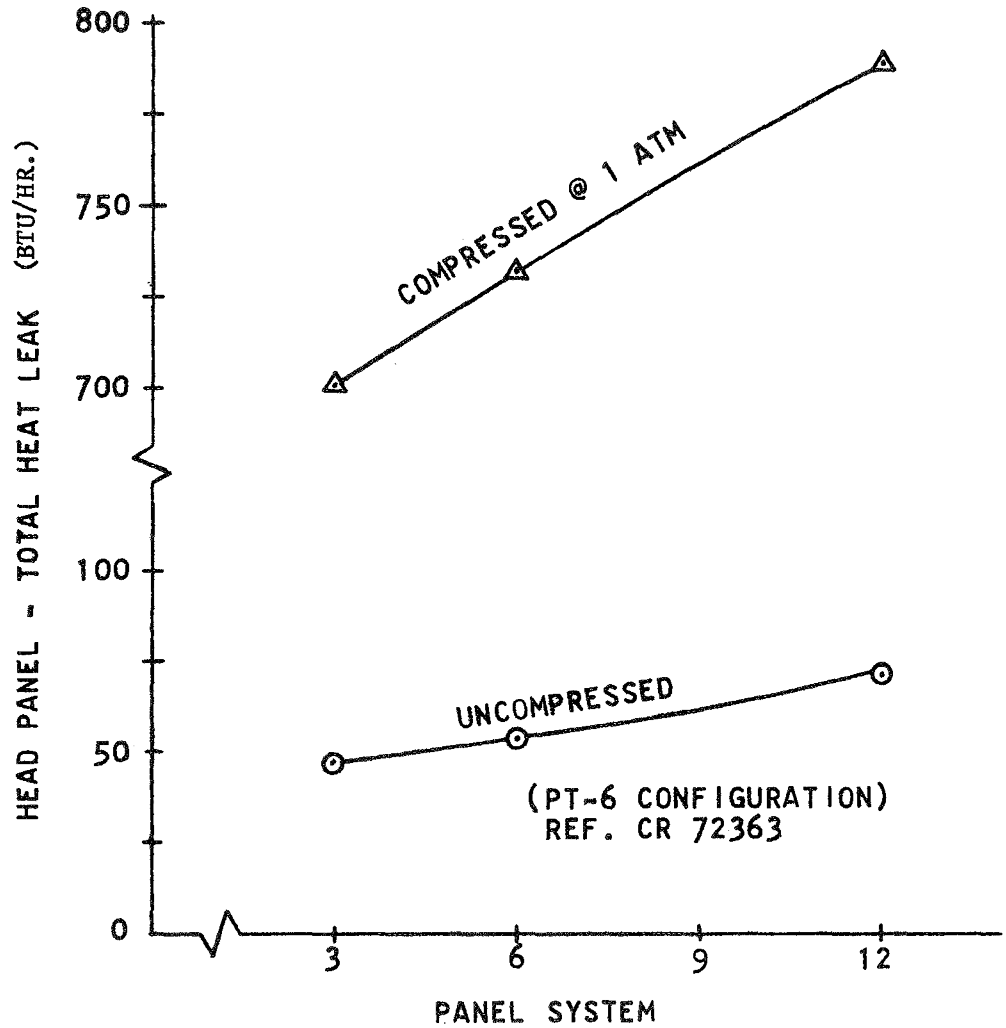


FIGURE 5

COMPUTER CALCULATED PERFORMANCE
HEAD PANELS - FULL SCALE INSULATION
TASK 1

TABLE 1

CALCULATED FULL SCALE INSULATION PERFORMANCE

	AVERAGE HEAT FLUX Btu/hr.ft^2 (Watts/M^2)		TOTAL HEAT LEAK Btu/hr. (Watts)	
	Compressed @ 1 atm.	Uncom-pressed	Compressed @ 1 atm.	Uncom-pressed
I. Computer Analysis [a]				
A. Best Performance				
(1) Circumferential-12 panel	7.05	.53	3177.4	236.9
(2) Head - 3 panel system	7.95	.54	701.4	47.1
TOTAL	7.2(22.7)	.53(1.67)	3879.8(1138.0)	284.0(83.3)
B. Realistic (shown SK-106295)				
(1) Circumferential-18 panel	7.30	.60	3286.2	270.0
(2) Head - 6 panel	8.28	.61	731.1	53.7
TOTAL	7.45(23.5)	.60(1.89)	4017.3(1177.0)	323.7(94.8)
C. Poorest Performance				
(1) Circumferential-18 panel	7.30	.60	3286.2	270.0
(2) Head - 12 panel	8.96	.82	791.5	72.5
TOTAL	7.58(23.9)	.64(2.01)	4077.7(1195.0)	342.5(100.4)
II. Flat Plate Data (NAS 3-7953) [b] (CR 72363 - pg.13) PT-6 Configuration				
TOTAL	6.47(20.4)	.13 (.41) @ .01 psi	3480(1020.0)	70.0(20.5)
III. Calorimeter (Ref. CR 72363)				
Total - Computer [a]	-	.45(1.42)	-	242.0(71.0)
Total - Test	10.0 [c] (31.5)	.63(1.97)	5384.0 [c] (1580.0)	339.5(99.5)

(a) Based on normal conductivity numbers computed from flat plate data.

(b) Flat plat data does not account for edge losses therefore numbers are the theoretical low limit for this insulation.

(c) Calorimeter data for the compressed case was not obtained at steady state conditions.

From the analysis, it was determined that panel size variations do not as greatly influence the performance as was expected. For the panel combinations tried, the range of thermal performance of the compressed cases varied by less than 5% while the uncompressed cases varied by less than 20%. For a combination of panels that are considered to be a realistic case, that is based on panels of a reasonable size, that can be readily fabricated, the total heat leak was degraded by 5% for the compressed case and 15% for the uncompressed case, as compared to the best system tried.

Refering to Figure 4 (sheet 3 of 4) it is noted that panels, Style GG and HH, are stepped along the length direction (L'). The reason for this is to allow the insulation panel to assume the desired temperature profile, thereby having a one layer insulation thickness near ambient, two layers at the intermediate section and finally achieving the full three layer system at the cryogenic temperature sections. Different length steps were run on the computer to determine the effect on insulation performance. The results indicate very little difference between steps of 3, 6 or 9 inches. Apparently, the thermal resistance approaches a limiting value in a relatively short length. This was demonstrated by changing the end dimensions of the 12 panel cylindrical system. The same results would be obtained for the 18 panel cylindrical system, and for the various head panel systems. It is, however, necessary to provide a stepped end insulation to establish a favorable temperature profile, to avoid an excessive heat leak.

4.1.3 Structural Analysis

The structural analysis of the insulation system was concerned with determining the effect of the launch environment on the panel casings, attachment points and foam spacers, and to devise a design to provide sufficient attachment and reinforcements to limit these stresses. The stresses considered are a result of a combination of thermal and mechanically induced loadings, as stated in the following summary. A summary of the results of the structural analysis of the full scale insulation system is included below. A complete write-up is included in Appendix 1.

1. Negligible thermal contraction (therefore stress) occurs in Mylar between 77°K and 20°K and, therefore, it is not necessary to perform the dynamic tests at Goddard using liquid hydrogen (Narmco adhesives, to be used in the panels, have previously been in 20°K. service with acceptable results.

2. The SEMI panels should be attached to the tank face and each other with VELCRO fasteners rather than adhesive, to allow for differential thermal contractions between tank and casings.

3. Lateral dynamic loadings of up to 5g and longitudinal combined acceleration and vibration levels of $\pm$ 8.5 g will result primarily in shear stresses on the order of 2.0 psi (1.38×10^{-4} newton per M^2) which are easily carried by VELCRO fastener design. Peel forces are expected to be negligible since the panels are not expected to buckle between VELCRO supports. To reduce the expected peel forces at the edge of the panels, the VELCRO fastener should be placed no closer than 2 inches (5.1×10^{-2}M) to the panel edge.

4. An attachment ratio of four square inches (2.4×10^{-3} M^2) of VELCRO closure per square foot (9×10^{-2} M^2) of panel is required to limit stress.

5. Buckling caused by longitudinal loads is not expected to fail the foam separators.

6. The panel casing seal stress is not expected to exceed 1.77 lb. per inch (3.1×10^2 newtons per meter) of seal, which is less than the peel strength of SEMI panel joints.

4.2 Model System

An objective of this program was to design and fabricate a sub scale model of the Full Scale Insulation System, including the design and fabrication of the test tank. Specifically, the SEMI system was designed for a 30 inch diameter (.76M) tank, having hemispherical heads, a 22 inch (.56M) long cylindrical center section and a simulated interstage skirt at one end. Overall length of the test tank including the support pipe was 70 inches (1.78M).

Although panel sizes, and attachments were scaled to evaluate the Full Scale Insulation, the SEMI system remains at 7 thermal spacers and 6 radiation shields enclosed within the 4 ply aluminized Mylar casing. An impermeable barrier of Mylar-Aluminum Foil-Mylar laminate was bonded to the air exposed surface of all panels to reduce air permeation into the panels and carbon dioxide out of the panels. The following subsections describe the various design, fabrication and tests of all Model System components.

4.2.1 Design

The following sections describe the efforts expended on interface scaling, panel layout, thermal analysis, and structural analysis of the panels, test tank, and related equipment.

4.2.1.1 Interface Scaling

The primary method and purpose of scaling the full size insulation system to produce a model system for test purposes was to note particularly the structural aspects, namely panel sizes, attachment points, and loadings due to thermal, mechanical or dynamic conditions, etc. The size of the test tank was not necessarily the deciding factor in scaling, as the tank was limited by physical constraints, i.e., test facilities and/or contractural requirements. The objective of this contract, considered the panel structural analysis to be of primary importance, with the thermal analysis being used to observe tradeoffs in the insulation system. The expected thermal performance of both the full size and the model system insulation designs were calculated. Thermal scaling was not a primary part of the work effort, but was performed with the understanding that the thermal performance could not be completely ignored. Structural performance could not be gained or improved at the expense of downgrading the thermal performance.

In view of this, certain similarities between the Model system and the Full Scale System have been observed.

1. Both systems involved a cylindrical tank with an insulated bulkhead and interstage skirt at one end.

2. Both systems would be insulated with the same materials, and the tankage is made of the same materials, and, therefore, the thermal contraction would be the same.

3. Both systems utilized three layers of shingled SEMI panels consisting of seven composite foam layers and six aluminized Mylar radiation shields in a four ply laminate of aluminized Mylar casing materials. (Same as employed in contract NAS3-7953)

4. Panels can be sized such that panel area-to-joint length ratios of the circumferential panels are similar for both systems. For example,

Full Size System			Model System		
System	Panel Size	Ratio $\frac{\text{Panel Area}}{\text{Joint Length}}$	System	Panel Size	Ratio $\frac{\text{Panel Area}}{\text{Joint Length}}$
12 Panel	90" x 96" (2.29M x 2.44M)	1.83	4 Panel	38" x 70.5" (.97M x 1.79M)	1.03
18 Panel	36" x 63" (.91M x 1.60M)	.95	6 Panel	38" x 48" (.97M x 1.22M)	.88

5. Observing from item 4, that a 18 panel system for the full size insulation results in panels of approximately the same size (36" x 63") (.91M x 1.6M) as the 4 panel model tank system (38" x 70.5") (.95M x 1.79M) it is possible to relate the VELCRO support area-to-SEMI panel area ratios, as well as making the distance between the VELCRO supports the same for the two systems.

Likewise, several dissimilarities between the two systems were noted.

1. Tank size - The Model system is approximately 1/4 size, as dictated by the physical constrainst, namely test facilities.

Full Scale System	10 ft. diameter x 20 ft. long (3.05M x 6.1M)
Model System	30 in. diameter x 68 in. long (.76M x 1.73M)

This results in a model system insulation length consisting basically of only two ends of the full scale insulation system, without the center section of panels. This compromise does not affect the structural requirements of the model system. Addition of the center section panels with subsequent smaller panel sizes would seriously degrade the thermal performance of the model system. The model tank diameter does provide a different panel curvature and, therefore, might possibly make the panels stronger due to this increased curvature over that of the full size insulation. This, however, is a compromise that must be made. The smaller diameter of the model system also effects the size of the polar panels, and the length of the insulated skirt, which decreases to 9 in (.23M) for the model tank as compared to a 34 in. (.86M) insulated skirt on the full size insulation.

2. Because of the different methods of tank support, there is no correlation on heat leaks via the piping or support skirt of the two systems.

In summary, therefore, the model system tank dimensions were dictated more as a result of physical testing restraints than as a result of scale down of the full scale insulation system. The overall length of the model system tester was limited to 72 inches at the NASA Plumbrook Station Facility, Sandusky, Ohio. Scaling the two systems for other than geometric correlation was not feasible. For example, no correlation existed between the heat leak of the two systems at either the skirt section or fill pipes. This is because the skirt section of the model system was not designed as a load carrying structure and, therefore resulted in a much lighter structure than the full size system. The inverse is true for the fill and drain lines since the model system was totally supported by a single centrally located pipe during thermal and dynamic testing as well as during transportation. Therefore, the model tank was sized by the physical restraints discussed above, while the insulation system was designed using panels of a similar size as used on the full scale insulation system, and in general following the same installation scheme as the full size insulation in regards to attachments, i.e. panel area to VELCRO attachment area ratios panel-to-panel joints, seals, etc.

Panel Layout - Model System Insulation

As a result of the scaling analysis, the model system used a four panel insulation system of 38 x 70.5" (.97M x 1.79M) panels which are similar in size to the full size insulation panels on the 18 panel system. The head panels (polar shingle panels) and the support skirt panels were not scaleable. Approximate panel sizes for the model system are listed in Figure 6.

4.2.1.2 Thermal Analysis

In order to predict the performance of the model system, the computer program written to evaluate thermal performance of the Full Scale insulation system was revised to calculate both ground hold and space conditions. An additional routine to cover the performance of the insulation system around the support head was added to the original program (See Appendix 2).

The predicted thermal performance of the model system is 340.0 (99.6) and 66.3 (19.4) Btu per hour (Watts) respectively for the compressed and recovered space conditions as presented in Table 2. For comparison purposes, model tank performance calculated using flat plate data and calorimeter data determined from Contract NAS3-7953 is also included in Table 2. As expected, the uncompressed data calculated from flat plate and calorimeter tests was much lower than the computer analysis of the model tank. This is probably attributable to the model tank configuration and is indicative of end condition penalties. It must be remembered that although these circumferential panels are of the same size (for structural purposes) as the 18 panel Full Scale Insulation, the overall thermal performance is not the same. In effect, the model tank thermally consists only of the two ends of the Full Size 18 panel insulation system. The accompaning thermal penalties when applied to a much smaller overall panel area, caused the average panel heat flux to be significantly higher. Likewise, the uncompressed inner skirt and polar panel calculated data was lower than the computed data for the model tank. This is due to the relative panel sizes, and subsequent large edge effect penalties which were included in the computer analysis but were not accountable in the calculations presented using the flat plate or calorimeter test data.

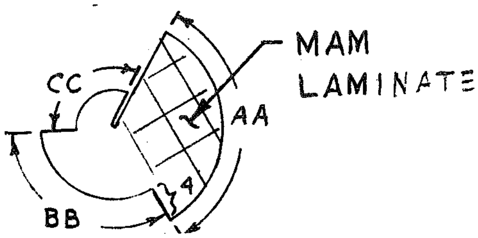
FIGURE 6

PANEL CONFIGURATION - MODEL SYSTEM INSULATION

TASK II - NAS 3-12045

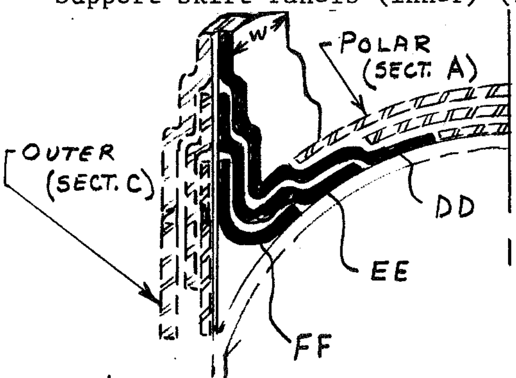
A. Polar Shingled Panels (3 Required)

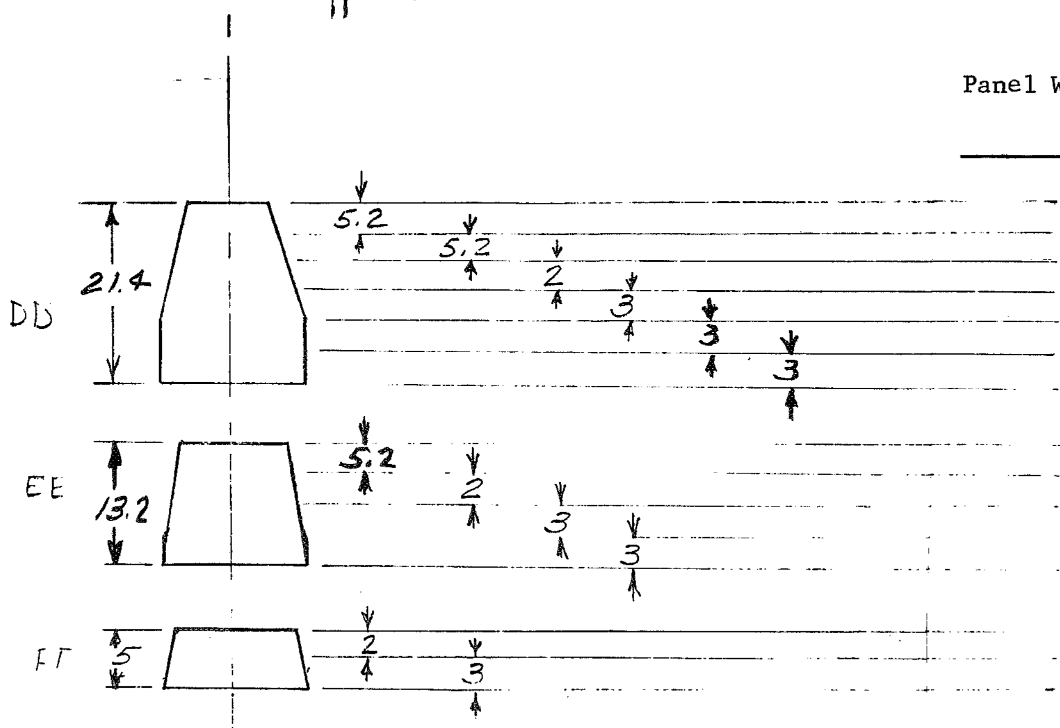
15-inch Spherical Radius



Panel Size (Inches)		
AA	BB	CC
28.2	20.7	10.1

B. Support Skirt Panels (Inner) (9 Required)



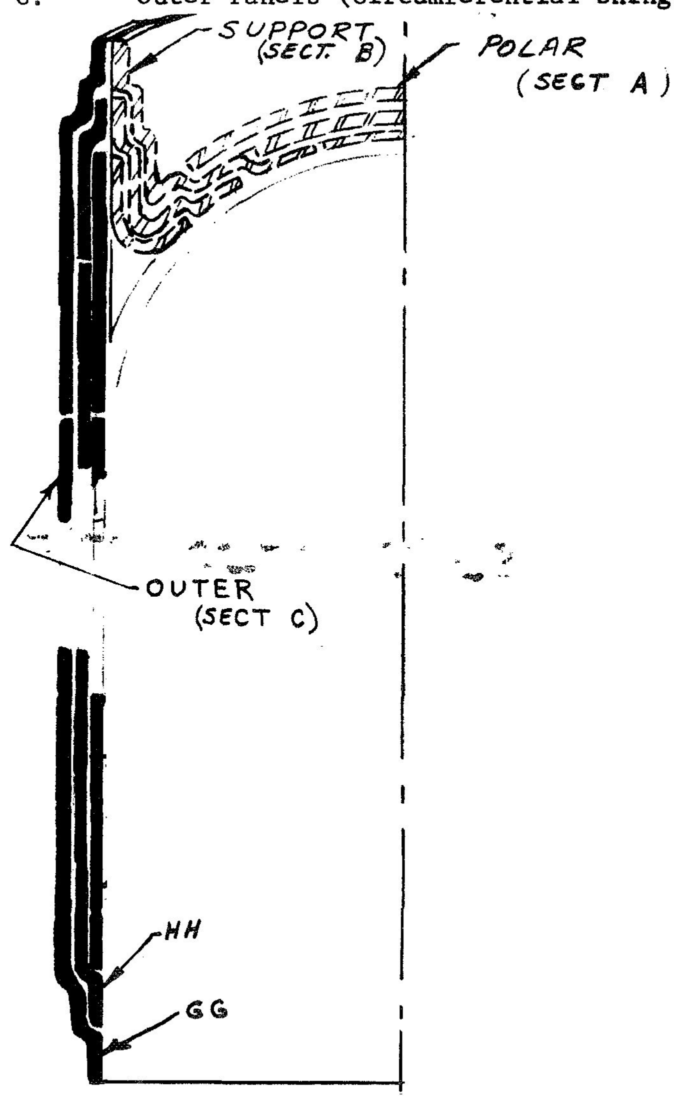


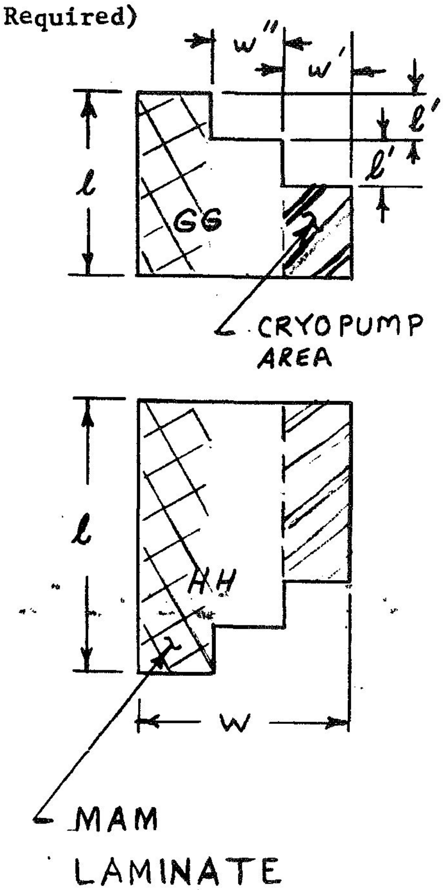
Panel Width - Inches

DD	10.1
	20.7
	28.2
	30.1
	31.4
	31.4
	31.4
EE	20.7
	28.2
	30.1
	31.4
	31.4
FF	28.2
	30.1
	31.4

FIGURE 6 (Cont'd.

C. Outer Panels (Circumferential Shingled) (8 Required)





Style	No. Required	Panel Size (Inches) l	w	Step l'	w'	w''
GG	4	19	75.8	3	24.4	25.4
HH	4	19	75.8	3	24.4	25.4

TABLE 2

CALCULATED MODEL SYSTEM THERMAL PERFORMANCE

		AVERAGE HEAT FLUX Btu/hr.ft^2 (Watts/M^2)		TOTAL HEAT LEAK Btu/hr. (Watts)	
		Compressed @ 1 atm.	Uncom-pressed	Compressed @ 1 atm.	Uncom-pressed
I.	Computer Analysis[a]				
	Inner Skirt & Polar Panels	10.5	2.5	69.1	16.5
	Circumferential Panels	9.9	1.3	253.9	32.8
	Total	10.1(31.9)	1.5(4.73)	323.0(94.7)	49.3(14.5)
II.	Flat Plate Data[b] (NAS 3-7953) (CR 72363 PT-6)				
	Inner Skirt, Polar Panel			42.4	.85
	& Circumferential Panels			164.9	3.3
	Total	6.47(20.4)	.13@ .01 psi (.41)	207.3(60.7)	4.15(1.22)
III.	Calorimeter Test Data NAS 3-7953 CR 72363 PT-6				
	Inner Skirt & Polar Panel			65.5	4.1
	Circumferential Panels			255.0 (c)	16.1
	Total	10.0[c](31.5)	.63(1.98)	320.5(94.0)	20.1(5.9)
IV.	Miscellaneous				
	Support Head Insulation[d]			13.8	13.8
	Support Skirt (Dummy - .037")[e]			3.2	3.2
V.	Expected System Performance				
	Inner Skirt and Polar Panels			69.1	16.5
	Circumferential Panels			253.9	32.8
	Support Head Insulation			13.8	13.8
	Support Skirt (Dummy - .037" Fiberglas)			3.2	3.2
	Total			340.0(99.5)	66.3(19.4)

(a) Based on normal conductivity numbers computed from flat plate data.

(b) Flat plate data does not account for edge losses therefore numbers are the theoretical low limit for this insulation.

(c) Calorimeter data for the compressed case was not obtained at steady-state conditions.

(d) Thermal conductivity-foam insulation in vacuum K = .002 Btu/hr.ft-°F

(e) Thermal conductivity reinforced fiber glass K = .2 Btu/hr.ft-°F

4.2.1.3 Structural Analysis

An objective of this program was to design a complete model tank test system having dynamic capabilities to sustain launch loads as detailed in Table 3 for the panels and test tank. The following sections detail the model test tank analysis and the insulation panel analysis.

4.2.1.3.1 Model Test Tank And Related Hardware

In order to evaluate the SEMI panel system for both thermal and dynamic conditions, a specially designed test tank was required. A layout for this tank, is shown in Figure 7. By contract, the tank must be between 28 (.71 to .91M) to 36 inches in overall diameter and a maximum of 6ft. (1.83M) overall length. The unguarded aluminum tank was supported from one head by a single stainless pipe connected to a bi-metalic transition joint and bolted flange. For thermal testing, the flange was attached to the Liquid Hydrogen Calorimeter Supply and Support Structure (NASA Plumbrook No. SK 68 1100). For dynamic testing, the bolted flange was attached to a Linde designed vibration table adaptor which mated with the vibration table of the Launch Profile Simulator (LPS) at Goddard. The cold guard supply structure was not needed for testing at Goddard since insulation thermal performance was not evaluated during dynamic testing. During transportation the bolted flange was attached to the shipping frame, thus providing a pendulum support for the SEMI panel insulated cryogenic test tank.

Per contract, the tank operating pressure ranged between 30 psi (2.07×10^5 newtons per M^2) and 75 psi (5.17×10^5 newtons per M^2), and the tank was capable of being evacuated. The various tank positions are depicted in Figure 8. The severity of the transportation conditions are expected to be less than the extremes of the test conditions with the exception of shock loading during transport, which are not expected to exceed 6g. Design parameters for the various tank loadings are listed in Table 4.

The fiberglass extension (simulated support skirt) was non-functional, i.e., not load bearing or structural supporting. Its function was to provide the geometric shape on which to attach the SEMI panels. This simulates the interstage support skirt on a typical multistage space vehicle.

A drawing of the tank SK-106298 (Figure 9) and details SK-106296, SK-106297 are included. Design calculations are included in Appendix 3. The tank mates with adapter SK-106294 (Figure 10) or the LeRC Cold Guard (SK-681100).

Goddard Model Tank Adapter

A short adapter section was designed and fabricated to locate the model tank within the accoustic liner of the Godard LPS Facility and to permit easy attachment to the vibration table. The design for this stand is shown on Figure 10(SK-106294). As can be seen, it consists of a straight section of tubing with flanges at either end to mate with the model tank (Ref. SK-106298) and an 18 inch (.46M) bolt circle at the Goddard Facility. 5 inch (.127M) diameter hand holes in the stand were provided to permit assembly of the model tank flange bolts. Fill and vent connections passed through the 5 inch (.127M) hole. See Appendix 4 for stress calculations.

TABLE 3

LAUNCH LOADS

(Table 1 Contract NAS 3-12045)

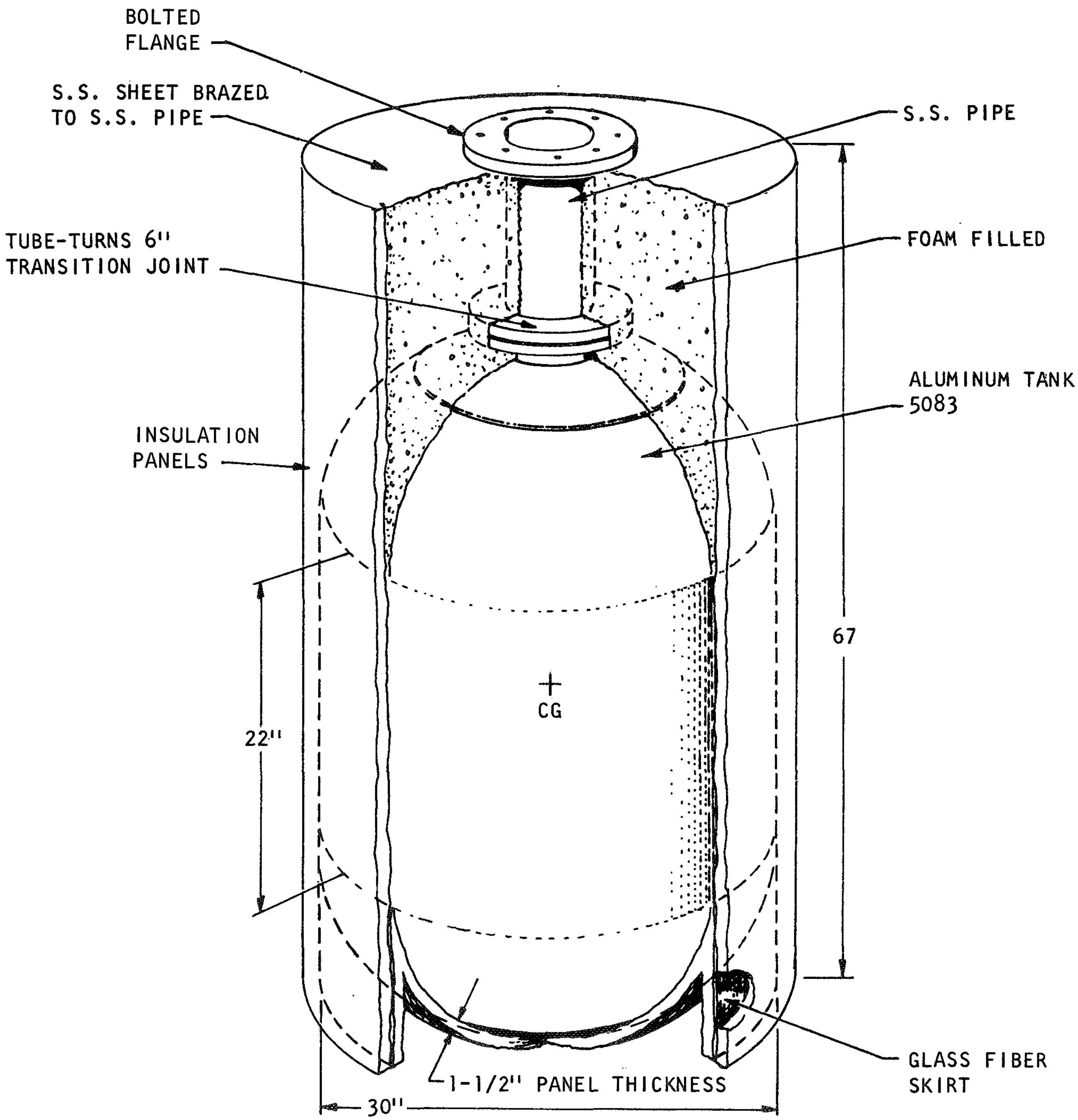
a) Longitudinal Load	-1.5 g's to + 6 g's
b) Lateral Load	1.5 g's (occurs when longitudinal load is + 2.5 g's)
c) Vibration	± 6g's vibratory longitudinal between 20 and 150 hertz (occurs when longitudinal load is 2.5 g's or less)
d) Acoustic	Overall sound pressure level (155 db)

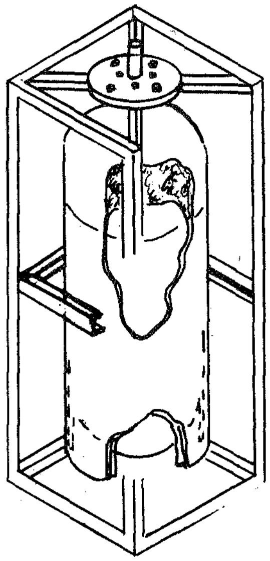
Octave Band Frequency (hertz)	Sound Pressure Level (db)
250	143
4000	138
8000	135

FIGURE 7

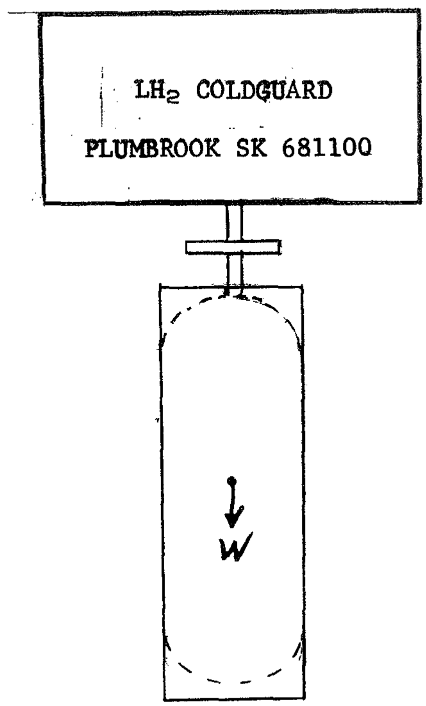
MODEL SYSTEM TEST TANK

CONTRACT NAS 3-12045

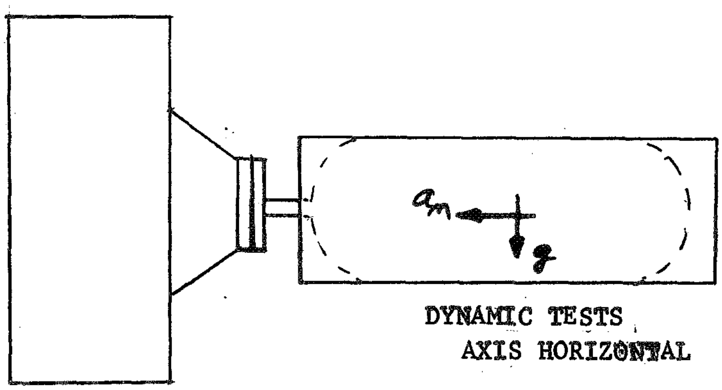




TRANSPORTING
AXIS VERTICAL



THERMAL TESTS
AXIS VERTICAL



DYNAMIC TESTS
AXIS HORIZONTAL

FIGURE 8

MODEL TANK POSITIONS

TABLE 4

MODEL TANK DESIGN PARAMETERS

A. GENERAL

1. Tank operating pressure range
   + 75 psid internal to 15 psid external

2. Temperature range
   ambient to -423°F.

3. Material 5083-0 aluminum

4. Hemispherical heads

5. Straight skirt section at end opposite of support heads (fiber glass)

6. Transition joint (Tube Turns. Ft. Type) between aluminum tank and stainless steel neck tube with flange.

7. All welded construction per ASME Unfired Pressure Vessel Code - Section VIII Division I & Welding Research Council Bulletin #107.

B. MODEL TANK DESIGN LOADS (CRITICAL)

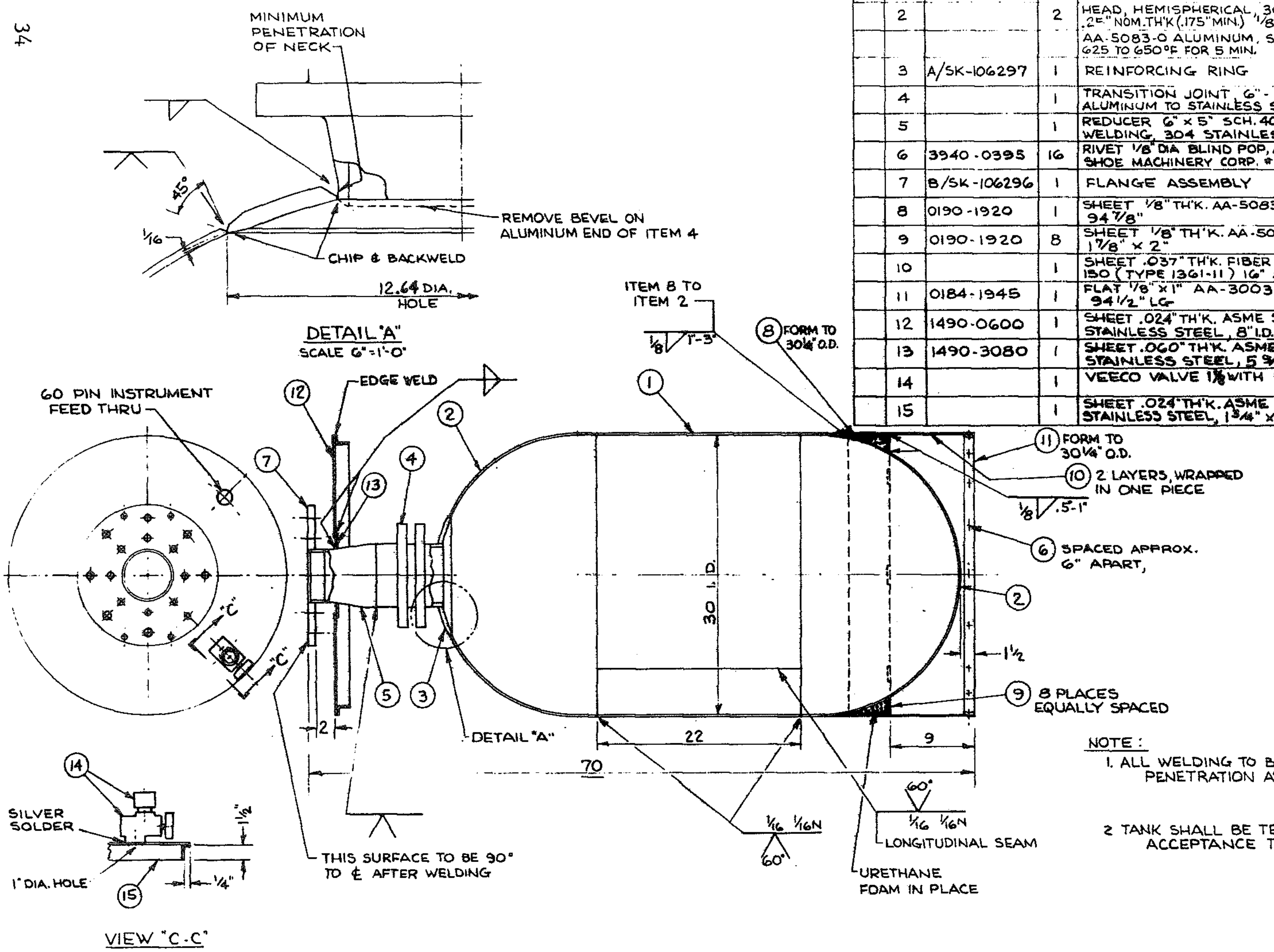
1.	Dynamic - Tank Horizontal	8.5 g on support tube in compression 1.5 g on support tube in bending 1.5 g on support tube in shear
2.	Special Condition - Tank Horizontal during LN_2 Cooldown	Bending and shear stress due to weight of LN_2 in tank during cooldown, i.e., longitudinal and lateral dynamic loadings equal zero.
3.	Thermal - Tank Vertical	Support tube in tension due to weight of tank including transition joints and tank filled with LN_2
4.	Transportation - Tank Vertical	Support tube in tension and bending due to 6 g shock loads (vertical and side loads) on empty tank

FIRST USED ON ASSEMBLY NO.

DIMENSIONAL TOLERANCES, FRACTIONAL ± ___" DECIMAL ± .01 ", ANGULAR ± 0 ° } UNLESS OTHERWISE NOTED

MACHINED SURFACES SHALL BE √

U/M	ITEM NO.	LINDE PART OR CODE NO.	REQ. FOR ONE	MATERIAL AND DESCRIPTION	DWG. ALT. LETT.
	1	0188-1014	1	PLATE, 1/4" TH'K. AA-5083-O ALUMINUM, 22" x 94 7/8"	
	2		2	HEAD, HEMISPHERICAL, 30" ± 1/16" I.D., 1" ST. FLG. .25" NOM. TH'K (.175" MIN.) 1/8" x 30° BEVEL EDGE, AA-5083-O ALUMINUM, STRESS RELIEVE AT 625 TO 650°F FOR 5 MIN.	
	3	A/SK-106297	1	REINFORCING RING	
	4		1	TRANSITION JOINT 6" - 150# FLANGED, ALUMINUM TO STAINLESS STEEL WELD NECKS	
	5		1	REDUCER 6" x 5" SCH. 40S, CONCENTRIC, WELDING, 304 STAINLESS STEEL	
	6	3940-0395	16	RIVET 1/8" DIA BLIND POP, ALUMINUM, UNITED SHOE MACHINERY CORP. # AD-44 H	
	7	B/SK-106296	1	FLANGE ASSEMBLY	
	8	0190-1920	1	SHEET 1/8" TH'K. AA-5083-O ALUMINUM. 5 1/2" x 94 7/8"	
	9	0190-1920	8	SHEET 1/8" TH'K. AA-5083-O ALUMINUM 1 7/8" x 2"	
	10		1	SHEET .037" TH'K. FIBER GLASS, FILON TYPE 150 (TYPE 1361-11) 16" x 192"	
	11	0184-1945	1	FLAT 1/8" x 1" AA-3003-H14 ALUMINUM. 94 1/2" LG.	
	12	1490-0600	1	SHEET .024" TH'K. ASME SA-240, TYPE 304 STAINLESS STEEL, 8" I.D. x 30" O.D. RING	
	13	1490-3080	1	SHEET .060" TH'K. ASME SA-240, TYPE 304 STAINLESS STEEL, 5 3/16" I.D. x 9" O.D. RING	
	14		1	VEECO VALVE 1 1/8 WITH 1 1/8 VEECO COUPLING	
	15		1	SHEET .024" TH'K. ASME SA-240, TYPE 304 STAINLESS STEEL, 1 3/4" x 92 5/8"	



NOTE :

1. ALL WELDING TO BE HELIARC AA-518 . OR ER-309 PENETRATION AS NOTED.

2 TANK SHALL BE TESTED PER MODEL TANK ACCEPTANCE TEST PLAN A/SK-106462

Sheet 1 of 3

FIGURE 9

LETT.	ALTERATION	DATE	BY	CH'K'D	APPV'D
A	GENERAL REVISIONS	4-23-70	REH		

TITLE: INSULATION TEST VESSEL NASA-LEWIS CONTRACT NAS3-12045

UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA NEW YORK

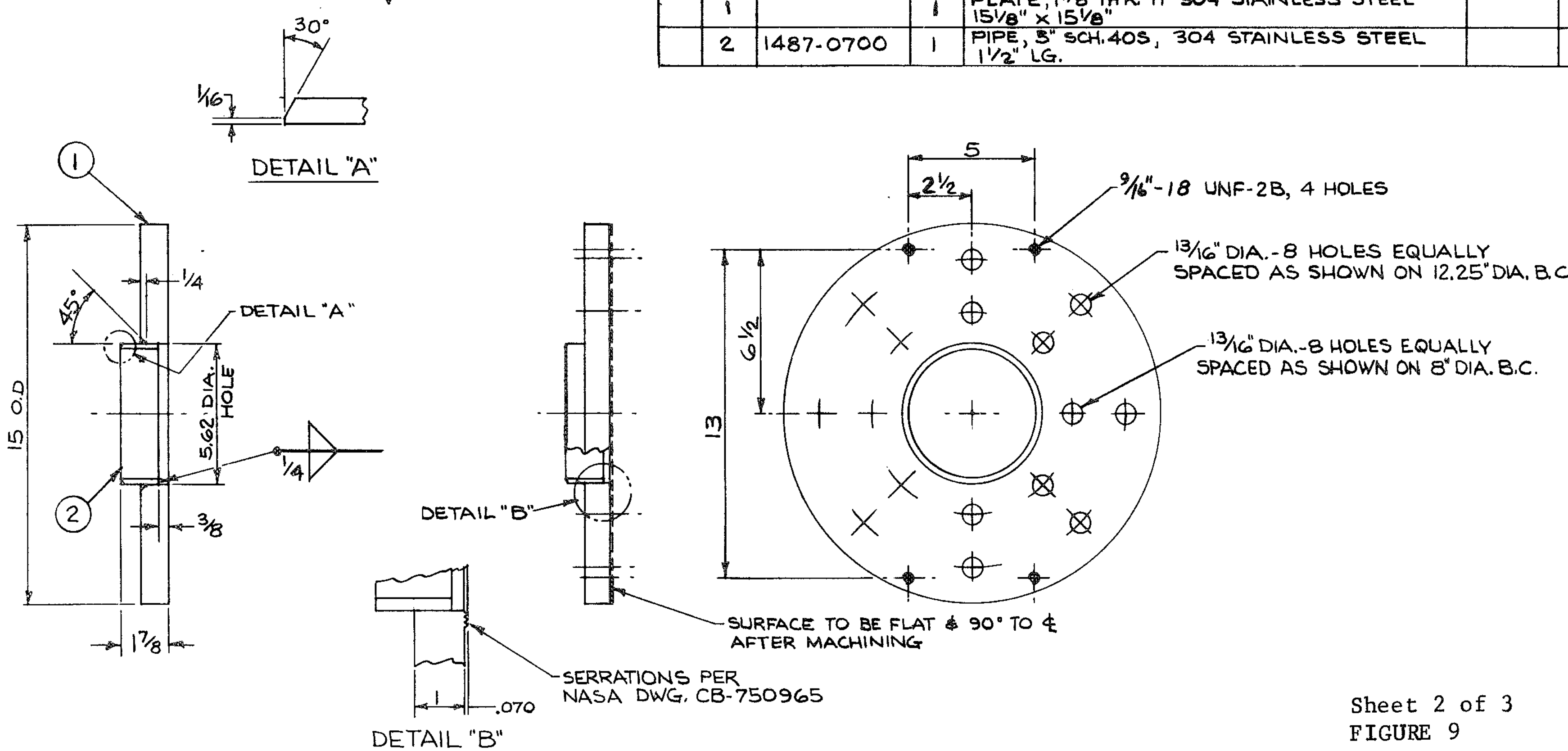
SIMILAR TO		
BY REH	DATE 10-1-69	SCALE 1/2" 1' 0"
CHECKED LKE	NO. SHTS. —	LATEST ALT. LETT.
APPROVED	SHT. NO. —	

C SK-106298

FIRST USED ON ASSEMBLY NO. C/SK-106298

TOLERANCES UNLESS OTHERWISE NOTED: FRACTIONAL ± 1/64" DECIMAL ±.005"
MACHINED SURFACES SHALL BE √

U/M	ITEM NO.	LINDE PART OR CODE NO.	REQ. FOR ONE	MATERIAL AND DESCRIPTION		DWG. ALT. LETT.
	1		1	PLATE, 1 1/8" TH'K. TP 304 STAINLESS STEEL 15 1/8" x 15 1/8"		
	2	1487-0700	1	PIPE, 5" SCH. 40S, 304 STAINLESS STEEL 1 1/2" LG.		



Sheet 2 of 3
FIGURE 9

TITLE: FLANGE ASSEMBLY INSULATION TEST VESSEL NASA-LEWIS CONTRACT NAS3-12045

UNION CARBIDE CORPORATION
LINDE DIVISION
TONAWANDA, NEW YORK

SIMILAR TO

BY	DATE	SCALE
REH	10-1-69	3"=1'-0"

CHECKED LXE — NO. SHTS. — LATEST ALT. LETT.
APPROVED SEM — SHT. NO. —

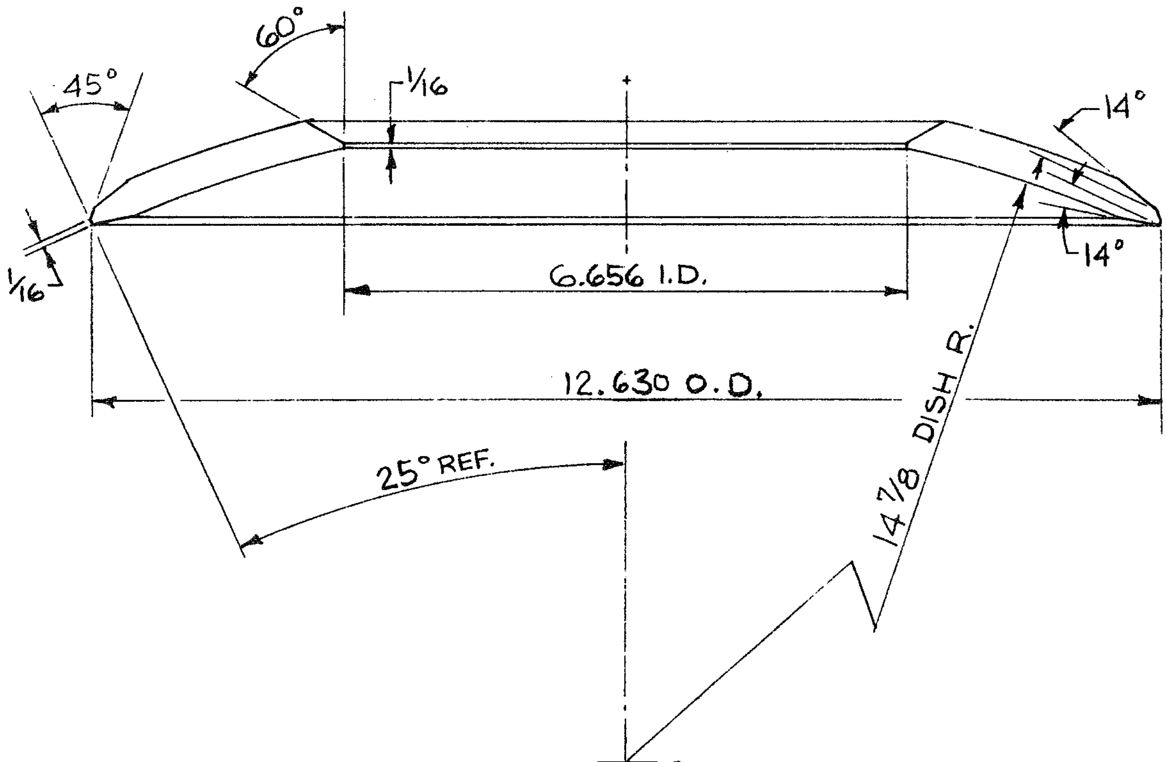
B SK-106296

LETT.	ALTERATION	DATE	BY	CH'K'D	APPV'D	LETT	ALTERATION	DATE	BY	CH'K'D	APPV'D

LETT.	ALTERATION	BY	CHK'D	DATE	APPV'D	LETT.	ALTERATION	BY	CHK'D	DATE	APPV'D

	ITEM NO.	PART OR CODE NO.	REQ. FOR ONE	MATERIAL AND DESCRIPTION			DWG. ALT. LETT.
	1		1	PLATE, 7/16" TH'K. AA-5083-O ALUMINUM 13" × 13"			

NOTE:
14 7/8" RADIUS TO BE PRESSED.



Sheet 3 of 3
FIGURE 9

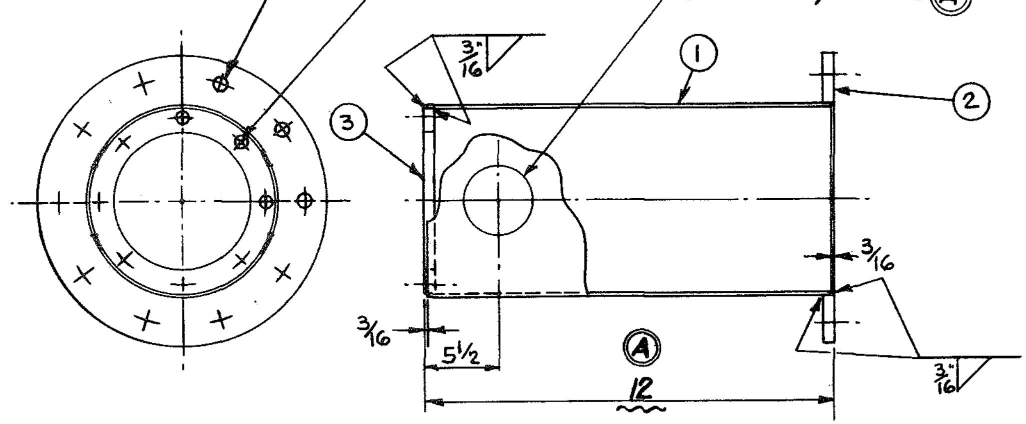
DIMENSIONAL TOLERANCES, FRACTIONAL ± 1/64" MACHINED SURFACES SHALL BE 125√ ± ___ % } UNLESS OTHERWISE NOTED
DECIMAL ± .010" ANGULAR ± ___ °

TITLE	WORK ORDER	FIRST USED ON	
REINFORCING RING, INSULATION TEST VESSEL, NASA-LEWIS, CONTRACT NAS3-12045	BY REH DATE 10-1-69	SCALE 6"=1'-0"	LATEST ALT. LETT.
	CHK'D LKE	SHEET —	SHEETS —
UNION CARBIDE CORPORATION LINDE DIVISION ENGINEERING DEPARTMENT TONAWANDA, NEW YORK	APPV'D [illegible]	A-SK-106297	

FIRST USED ON ASSEMBLY NO. B

TOLERANCES UNLESS OTHERWISE NOTED:
.X = ±.1, .XX = ±.06, ANGULAR ± °
MACHINED SURFACES SHALL BE √
FRACTIONAL ± 1/64

U/M	ITEM NO.	LINDE PART OR CODE NO.	REQ. FOR ONE	MATERIAL AND DESCRIPTION		DWG. ALT. LETT.
	1	0187-4642	1	TUBE 14" O.D. × 3/16" WALL, AA-5083-O, ALUM. 11 5/8" LG.		A
	2	0188-2527	1	PLATE 3/4" TH'K, 21" O.D. × 14 1/16" I.D., RING, AA-5083-O ALUMINUM		
	3	0188-2527	1	PLATE 3/4" TH'K. 13 9/16" O.D. × 10" I.D. RING, AA-5083-O ALUMINUM		



NOTE :
1. ALL WELDING TO BE HELIARC, AA-5183
2. END SURFACES MUST BE PARALLEL AND FLAT WITHIN .02"

FIGURE 10

LETT	ALTERATION	DATE	BY	CH'K'D	APPV'D
A	BM ITEM 1: 11 5/8" DIM. WAS 29 5/8"; ON DWG: 12" DIM. WAS 30"; 5" DIA. HOLE WAS 1 REQ'D.	3-18-70	F.S.	J.SM	SSM 3/19/70

TITLE:
ADAPTER, VIBRATION TABLE, NASA-LEWIS CONTRACT NAS3-12045

UNION CARBIDE CORPORATION
LINDE DIVISION
TONAWANDA, NEW YORK

SIMILAR TO

BY	DATE	SCALE
REH	9-30-69	1 1/2" = 1'-0"
CHECKED SCA		NO. SHTS. —
APPROVED SSM	10/6/69	SHT. NO. —

LATEST ALT. LETT. A

B SK-106294

Goddard Flange Assembly (Vibration Test)

In order to perform the vibration test at cryogenic temperatures a special flange assembly was designed to provide a fill and drain capability with the tank in the horizontal position (Figure 11 Item 2 C/SK 106415). Two safety relief (pressure burst disks) were included. The fill tube was designed to provide an interference fit of 0.4 inch (0.1M) with the side of the tank, i.e. loaded cantilever beam. This provided the necessary force to assure that the fill pipe remained in intimate contact with the tank throughout the vibration test without rubbing.

Model Tank Transporter (Handling Fixture)

Since the model tank must be supported entirely from one end in various positions and attitudes during test and transportation, a special handling fixture or transporter was required. The transporter is shown as Figure 12 (SK-106293). Special requirements for the transporter in addition to supporting the tank vertically for transportation are as follows:

1. Allow tank to be mated to the thermal tester at Plumbrook (SK-681100) while the tank is still supported by the transporter.

2. Allow the tank (and transporter) to be rotated about a horizontal axis and mate with the Goddard "acceleration and vibration end cap" prior to being separated from the transporter.

The transporter was designed to sustain a 6g side loading, with the tank installed. The tank was rigidly bolted to the transporter head structure, without shock attenuation supports of any kind. The transporter was faced with 1/2 inch (1.27×10^{-2}M) thick plywood to provide protection for the model tank during shipment.

Stress calculations for the design of the transporter are contained in Appendix 5.

4.2.1.3.2 Insulation Panel

Because of similarities in panel size between the Model System and the Full Scale System insulation, an additional evaluation of the Insulation panel structure was not required. The panels were attached to the tank and to each other using VELCRO, at a ratio of 4 square inches ($2 \times 10^{-3}M^2$) of VELCRO attachment for each square foot ($.09M^2$) of panel, with the attachments placed no closer than two inches (.05M) to the panel edge as recommended in Section 4.1.3. VELCRO placement was as indicated on Figure 13 & 14.

For panel seal-off, a small O-ring sealed unit was designed to allow entrance to the panel for evacuation and carbon dioxide backfilling and/or placement and removal of a vacuum gauge. Neither operation will require cutting of the SEMI panel casing. The seal-off device and related adapters are shown on Figure 15. Operation is as follows. To enter or exit the panel, a simple purge bag is attached to the panel to surround the seal-off area, and purged with CO_2

LETT.	ALTERATION	DATE	BY	CH'K'D	APV'D.
F	GENERAL REVISION				

U/M	ITEM NO	PART OR CODE NO	REQ FOR ONE	MATERIAL AND DESCRIPTION
LB	1	1488 - 6165	1	PL 1.00 × 9.62 × 9.62 / 304 STN STL
IN	2	1487 - 4370	1	TUBE .75 OD × 51.00 / 304 STN STL .065 WALL
IN	3	1487 - 4750	1	TUBE 1.00 OD × 16.00 / 304 STN STL .065 WALL
	4		2	SAFETY HD, SCR TYPE, .50 / BS & B TYPE SA 8 AISI 303 S.S. INLET, STL OUTLET
	5		2	RUPTURE DISC, .50 @ 100 PSI / FOR BS & B SAFETY HD ALUMINUM

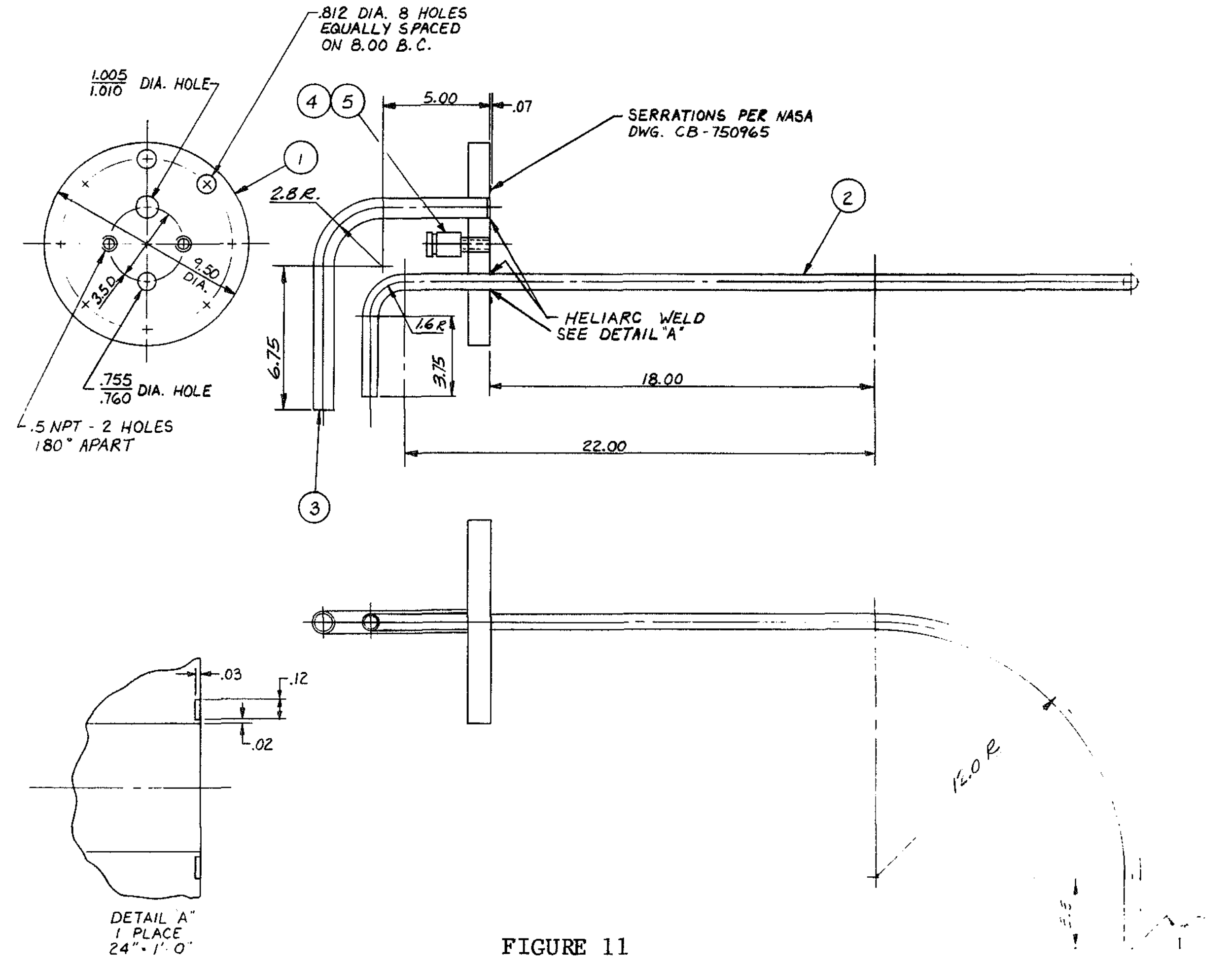


FIGURE 11

TITLE

FLANGE ASSEMBLY VIBRATION TEST
NASA LERC CONTRACT NAS3-12045

BY	DATE	CHK'D	SCALE
RTP	10-23-69		3 = 1
APPV'D		SHEET	SHEETS

UNION CARBIDE LINDE DIVISION
ENGINEERING DEPARTMENT
TONAWANDA, NEW YORK

C/SK-106415

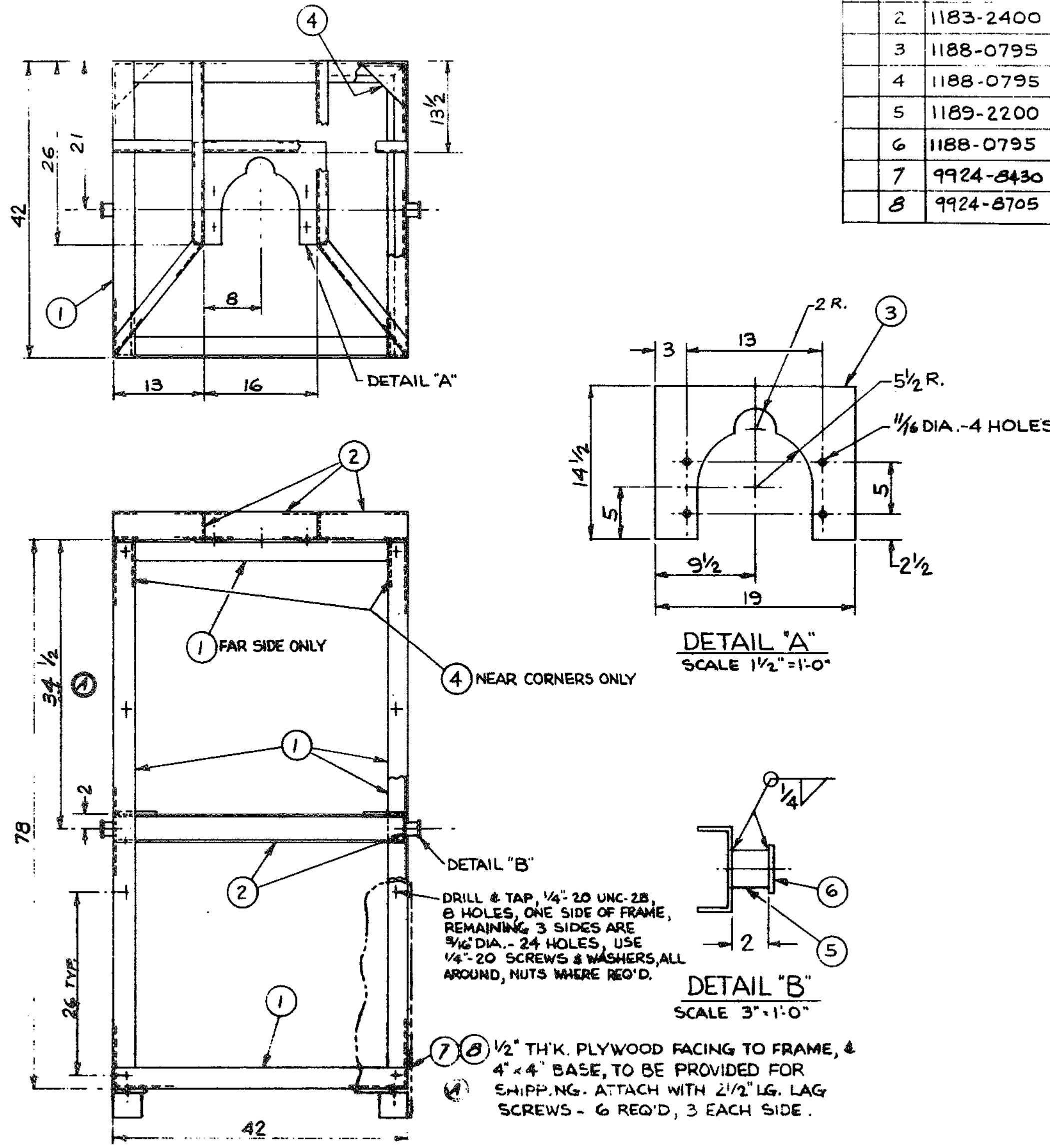
C/5K-106293

40

U/M	ITEM NO.	LINDE PART OR CODE NO.	REQ. FOR ONE	MATERIAL AND DESCRIPTION		ALT. LETT.
	1	1180-3440	49 FT	ANGLE 3 x 3" x 5/16, STEEL		
	2	1183-2400	22 FT	CHANNEL 4" x 1 5/8" @ 5.4 # STEEL		
	3	1188-0795	1	PLATE 1/4" TH'K. STEEL, 14 1/2" x 19"		
	4	1188-0795	1	PLATE 1/4" TH'K. STEEL, 6" x 12 1/2" (MAKES 4)		
	5	1189-2200	2	ROUND, 1 3/4" DIA. STEEL, 2" LG.		
	6	1188-0795	2	PLATE 1/4" TH'K. STEEL, 2 1/4" x 2 1/4"		
	7	9924-8430	4	SHEET, 1/2 TH'K, PLYWOOD, 42" X 78"		A
	8	9924-8705	2	TIMBER, 4X4X42, SPRUCE		A

FIRST USED ON ASSEMBLY NO.

DIMENSIONAL TOLERANCES, FRACTIONAL ± ___" DECIMAL ± ___", ANGULAR ± ___° MACHINED SURFACES SHALL BE √ } UNLESS OTHERWISE NOTED



NOTES:

1. ALL CONSTRUCTION TO BE METAL ARC WELD, FULL FILLET ALL AROUND, WHERE POSSIBLE.
2. PAINT ALL METAL (PRIME COAT ONLY) PER SPI-49.200, METHOD C-2
3. PRIME & PAINT WOOD WITH SUITABLE EXTERIOR GRADE PAINT.

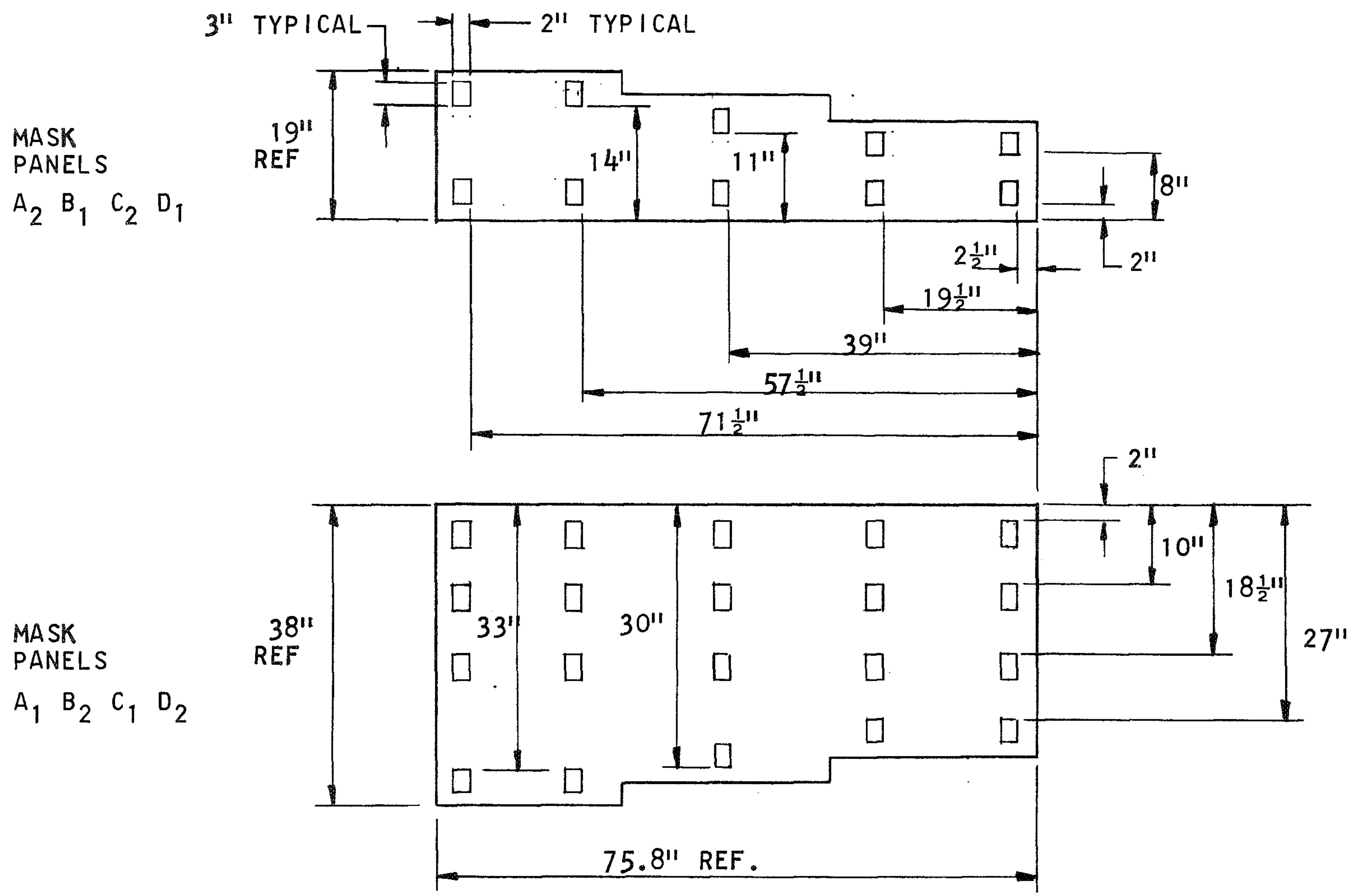
FIGURE 12

TITLE: MODEL TANK TRANSPORTER NASA LEWIS, CONTRACT NAS3-12045

SIMILAR TO | BY REH | DATE 9-30-69 | SCALE 1" = 1'-0" | CHECKED | APPROVED | C SK-106293

UNION CARBIDE CORPORATION
LINDE DIVISION
TONAWANDA, NEW YORK

LETT.	ALTERATION	DATE	BY	CHK'D	APPV'D
A	[illegible]	3/19/70	[illegible]		



VELCRO STYLE #65 CLOSURE W/SA 0145A COATING

FIGURE 13 VELCRO PLACEMENT, CIRCUMFERENTIAL PANELS (BACKSIDE)
(SEE FIG. 6 FOR PANEL SIZES)

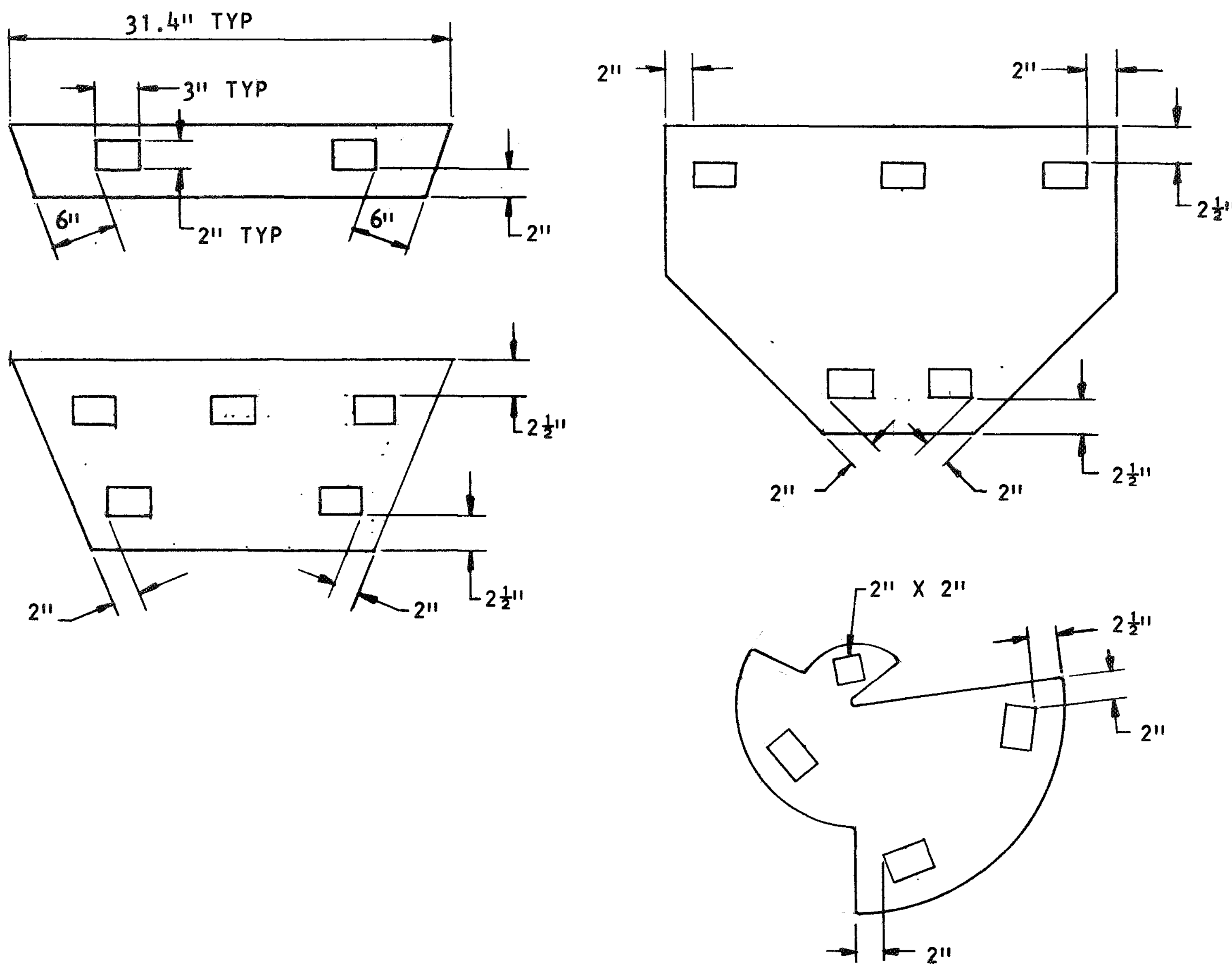
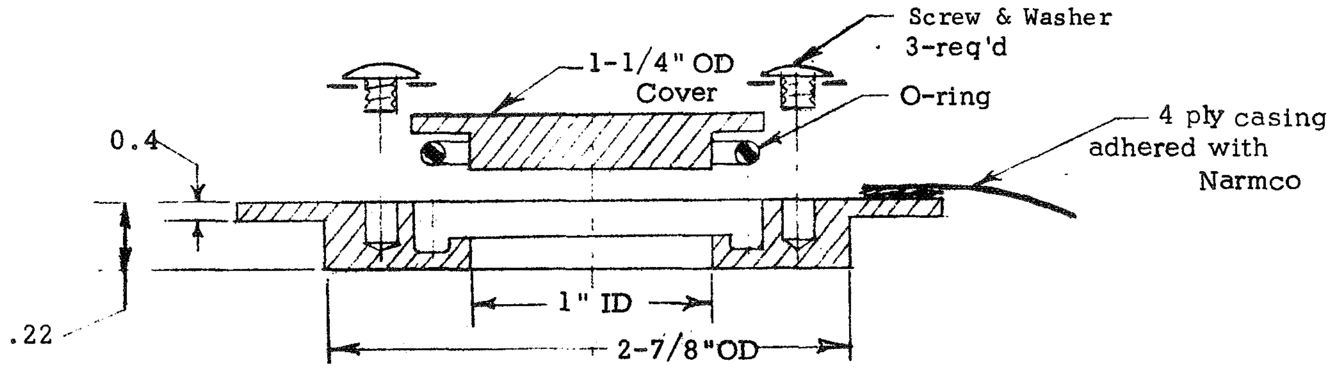
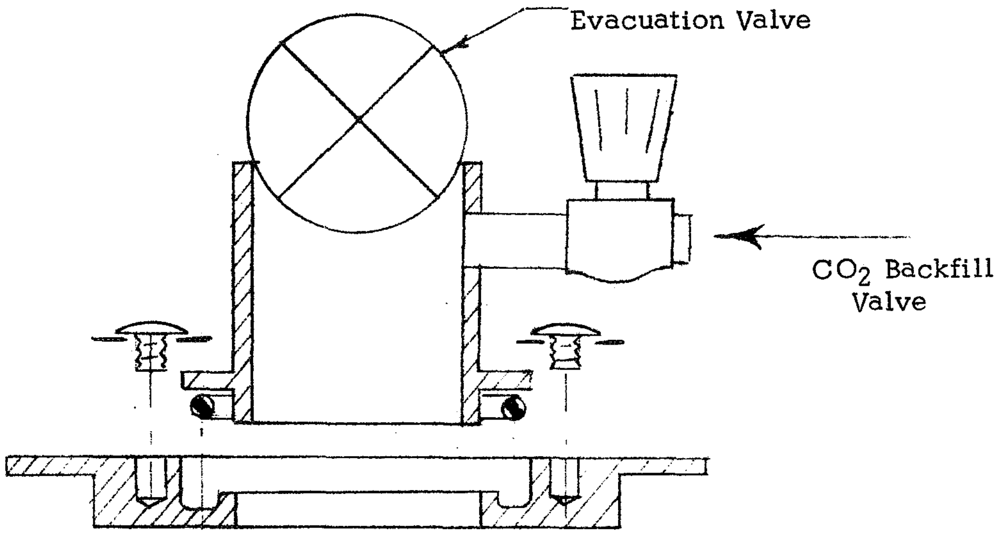


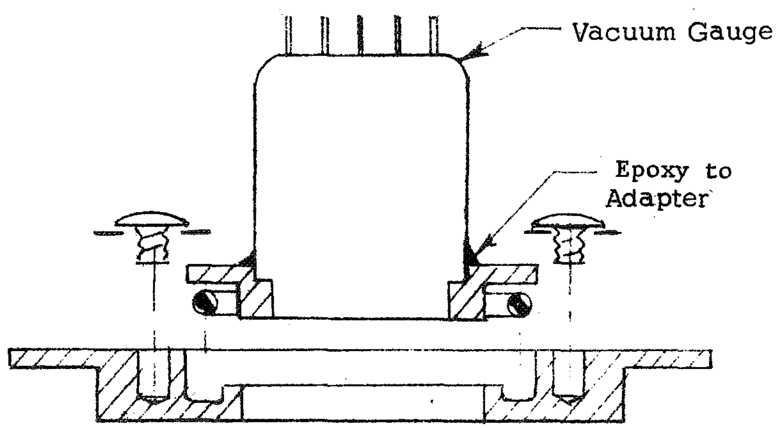
FIGURE 14 VELCRO PLACEMENT, INNER SKIRT AND POLAR PANELS (BACKSIDE)
(SEE FIG. 6 FOR PANEL SIZE)



Seal-Off Device - Exploded View



Evacuation Adapter



Gauge Adapter

FIGURE 15

Panel Seal-Off Device and Related Adapters several times before the desired change is made. Since these transfers occur with the panel at ambient pressure, system contamination, if any, consists only of gas diffusion between the panel and the purge bag. This amount of contamination if present would be insignificant. This seal-off device with the associated equipment permits evaluation of the CO_2 gas in the panel without necessitating re-evacuation and subsequent backfill. A Viton-A O-ring was used to seal the panel. The complete seal-off device is fabricated of aluminum, weighs 32 grams, and therefore did not require external supports during vibration tests.

4.2.2 Fabrication

4.2.2.1 Tank Assembly and Related Hardware

The tank (C/SK 106290) was assembled using a semi-automatic welding procedure with a hand held heliarc welding torch and a rotating positioner. The 5083 aluminum material was welded with AA-5183 filler rod, and the 304 stainless was welded using ER 309 filler rod, as per Figure 9 drawing C/SK 106298 (see Section 4.2.1.3.1).

The dissimilar metal transition joint-to-upper head reinforcing ring weld was 100% radiologically examined and determined to be structurally sound. The transition joint/ring weld was then leak checked on Vecco Model MS-9 helium leak detector. A steady state leak rate of 3.2×10^{-8} atm. cc air/sec. was determined for the test set up, which because of the shape, included two flat rubber seals. This rate was determined after a 3 hour settle out, and indications are that it was likely permeation through the rubber (temporaty seal) rather than leakage at the weld area.

After tank assembly, the initial helium leak check disclosed a small leak in the upper head to cylinder weld. The leak was repaired by grinding and then rewelding a section approximately four (4) inches (.1M) in length. The final leak rate for the tank was 1×10^{-10} atm. cc air/sec. excluding the leakage through the aluminum gasket. (A tank leak rate of 1×10^{-7} atm. cc air/sec. or less is acceptable.)

Vibration Adapter (B/SK 106294)

The adapter was fabricated of 5083 aluminum alloy and AA-5183 filler rod. The unit was spot x-rayed after fabrication to assure weld structural soundness. Both flanges were then machined to obtain parallel and flat surfaces.

Transporter-(C/SK 106293)

The transporter frame was fabricated of commercially available structural steel shapes welded together by a metal arc process. The transporter frame with the test tank installed is shown in Figure 16. The frame is covered with plywood for shipping.

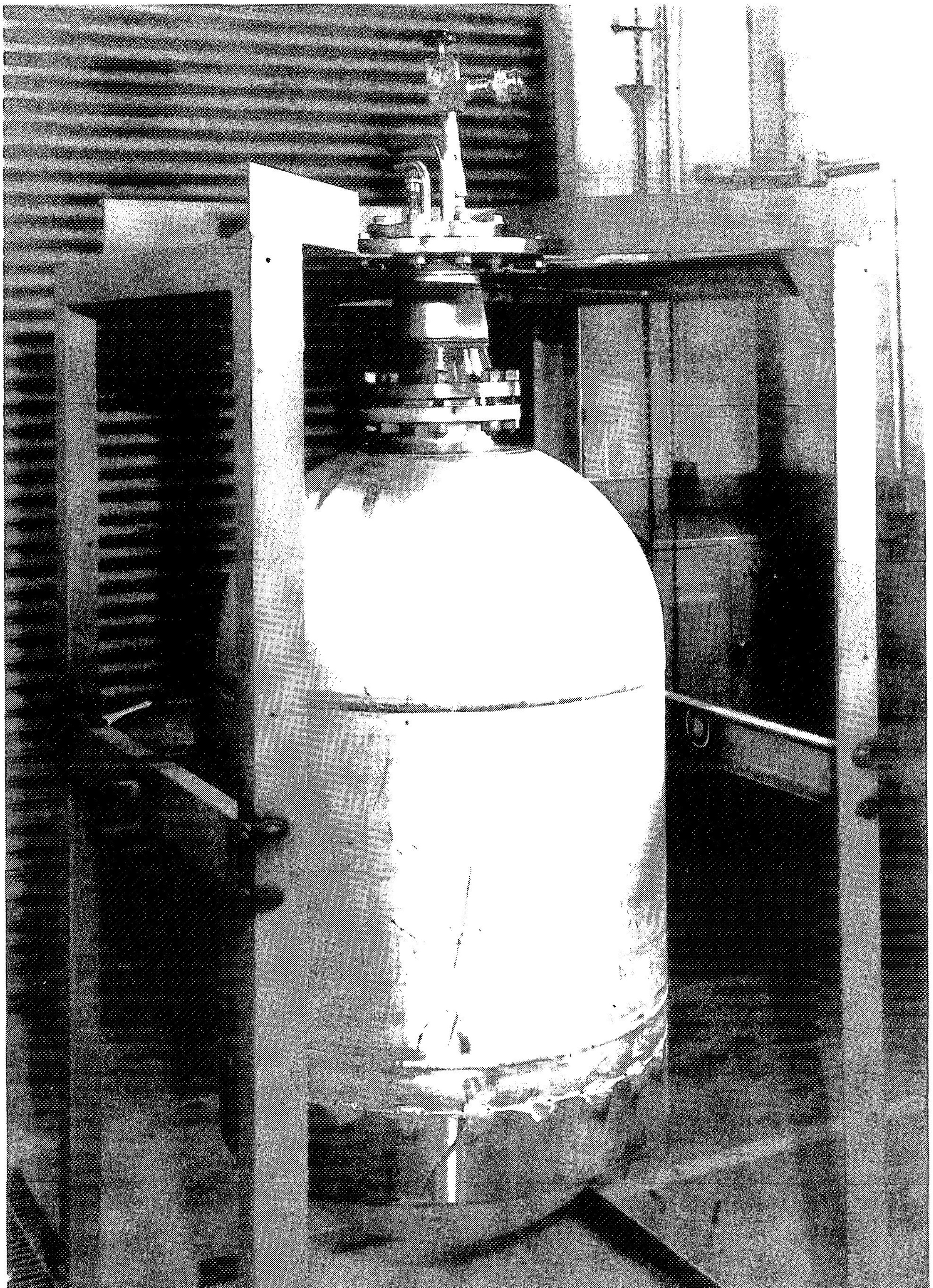


FIGURE 16. Transporter C/SK 106293 With Test Tank C/SK 106298

4.2.2.2 Panel Fabrication

The SEMI panels were fabricated using techniques developed under previous UCC contracts NAS 3-6289 (Ref. 1) and NAS 3-7953 (Ref. 2). The panels consist of a casing of 4 ply aluminized Mylar, utilizing Narmco adhesive joints, and filled with a multilayer insulation consisting of open cell rigid polyurethane foam and aluminized Mylar radiation shields. Vacuum tight casings were fabricated for the circumferential and polar panels, however, such was not the case for the inner skirt panel casings as will be discussed later. Panel fabrication includes several operations which are listed below and discussed separately in the following section.

1. Preprocessing the aluminized Mylar radiation shields by heating in air for 24 hours at 150°F. (338°K)

2. Punching the foam spacers including vacuum cleaning of the foam to remove the foam dust generated during the bun slicing operation.

3. Vacuum forming both casings to obtain the required pleats and recovered panel thickness without residual compression.

4. Getter installation, adhesive application (Narmco 7343/7139 preceded by a prime coat of Goodyear G-207 solution), with pressure cure.

5. Panel evacuation and helium leak checking.

Results of development work performed under a previous SEMI panel contract (NAS3-6289) indicated that only the aluminized Mylar radiation shield needed to be pre-conditioned. It was determined that exposing the radiation shields to warm circulating air for 24 hours would sufficiently remove the hydrogen, and that subsequent vacuum pumping of the completed panel at ambient temperatures would be adequate to remove contaminants. The rest of the panel materials were determined to benefit very little as a result of any preconditioning.

Each panel contains 7 spacers and six radiation shields. The spacer used for the panels was a three layer foam composite consisting of two 0.02 inch (5.08×10^{-4}M) thick open cell rigid polyurethane foam layers containing punched 1 1/4 inch (3.2×10^{-2}M) square holes on 2 inch (5.1×10^{-2}M) centers, and one 0.02 inch (5.08×10^{-4}M) thick layer of unpunched open cell rigid polyurethane foam. The two punched hole layers were positioned relative to each other such that support was achieved only at the intersection of the two webs, as shown in Figure 17. Several single layers of punched foam are shown in Figure 18. Also apparent in this figure is a bar containing a row of 21 cutters (developed under contract NAS3-7953), which was traversed along the punching table. Indexing pins locate each row of punched holes. With this method, seven layers of foam could be punched satisfactorily at one time.

Panel casings were constructed of a composite casing material consisting of impermeable Mylar/aluminum/Mylar (MAM) and 4-ply laminate for the outer air exposed section and 4-ply aluminized Mylar laminate for the

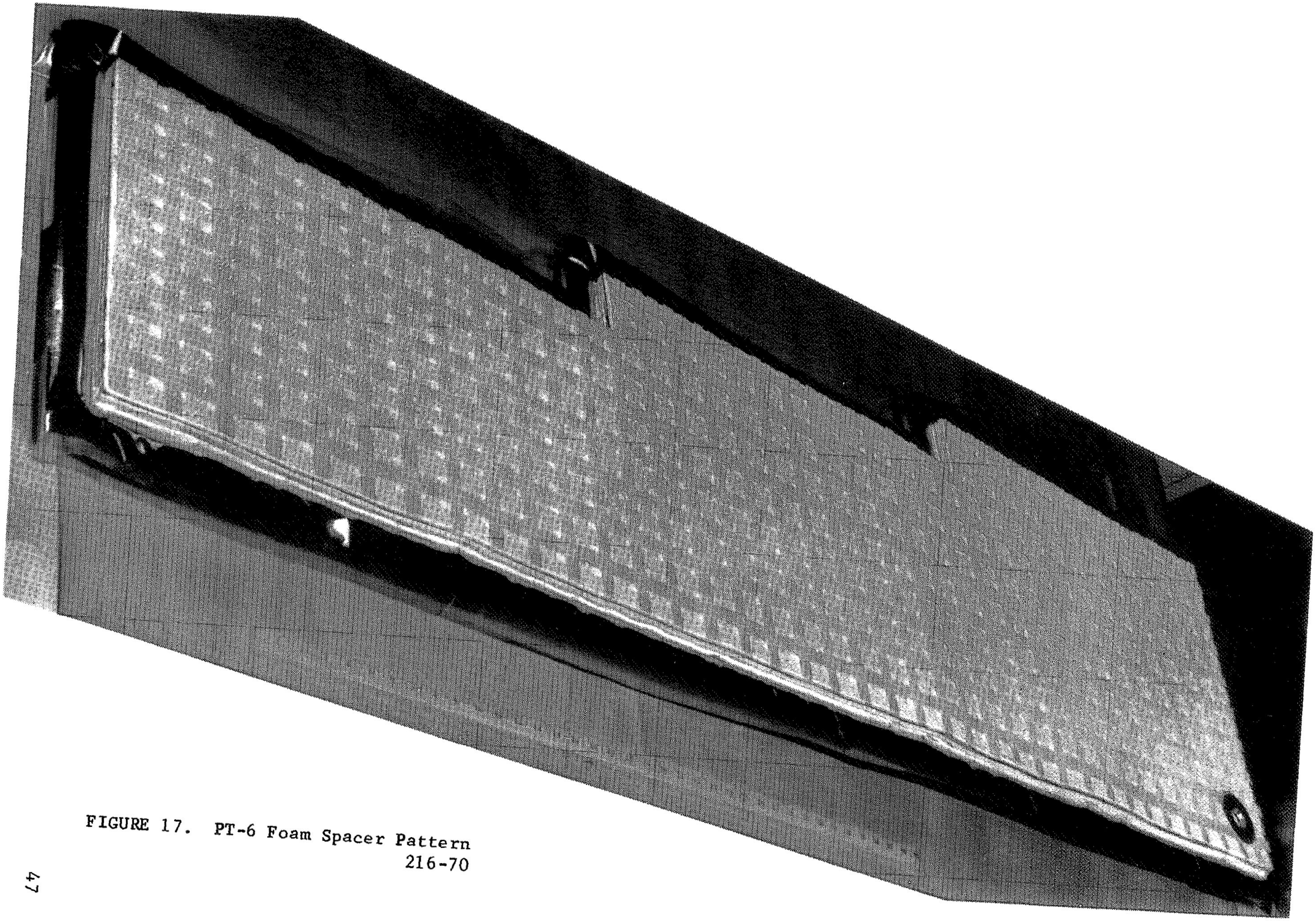


FIGURE 17. PT-6 Foam Spacer Pattern
216-70

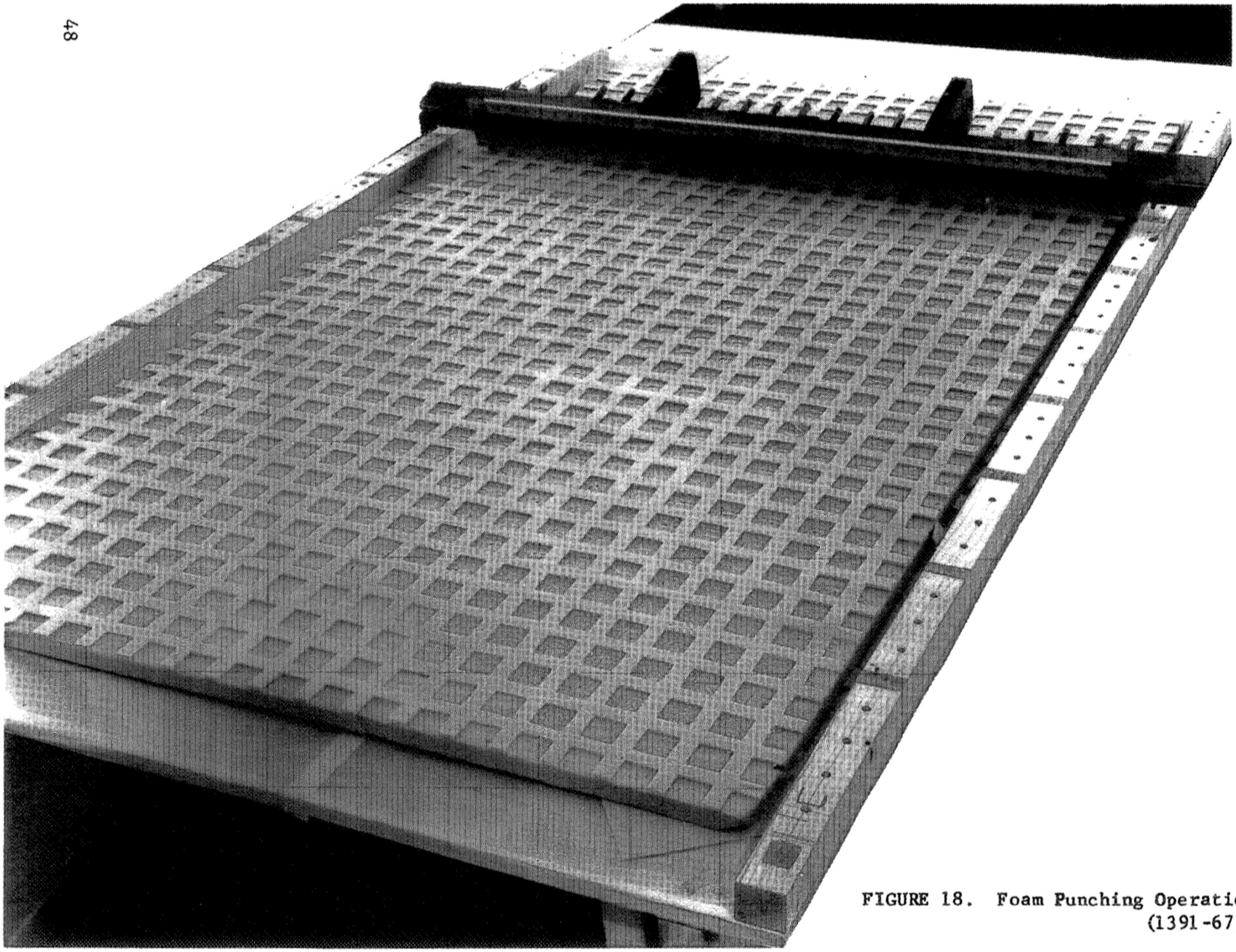


FIGURE 18. Foam Punching Operation (1391-67)

remaining 5/6 of the panel area. (See Section 4.2.1.2 Figure 6.) This combination was designed to achieve a casing with a highly impermeable surface that is exposed to air, yet allowing the remainder of the panel to exhibit a low thermal conductivity to enhance thermal performance.

One 3/4 inch (1.88×10^{-2}M) diameter hole was cut in each of the three bottom shields in the area immediately under the evacuation plate, as an aid in evacuating the panels. The holes, located on a one inch (2.54×10^{-2}M) diameter circle, were indexed so as to be located 120° from the hole in the adjacent shield. A 1-13/16 inch (4.60×10^{-2}M) diameter hole was cut in the foam layers and top three shields to accomodate the evacuation plate. This permitted the plate to be flush with the outside face of the panel.

The MAM was laminated to the 4-ply casing using Goodyear 207 prime coat. Both surfaces were coated and allowed to air dry for $\sim$24 hours. After the drying period, the materials were assembled and hand rolled with a teflon roller, while being heated sparingly with a hand held heat gun. After bonding the two materials together with Goodyear G-207 prime, the composite casing was vacuum formed (stretched) to allow for the insulation thickness. Three-eights inch (9.67×10^{-2}M) centers were formed in the casing at the same time, to allow for additional material to account for the difference in diameter as the panels were curved around the tank.

Prior to panel assembly, one gram of hydrogen getter was placed in the warm side foam spacer in the immediate area surrounding the evacuation manifold.

Panel adhesive joints were made in the following manner: After degreasing the Mylar bond surfaces with Methyl Ethyl Ketone (MEK) and allowing the casing to air dry, both surfaces were primed with a solution of G-207 formulation. The prime consisted of the following:

Goodyear 207 B	100 gms
Toluene	63 gms
MEK	27 gms
Goodyear 207 C	4 gms

After the prime had cured for 24 hrs. at room temperature, Narmco 7343/7139 adhesive was applied by brush to both Mylar surfaces. Narmco adhesive is a 2 part system consisting of a resin (7343) and a curing agent (7139). The curing agent is supplied in small pellets which must be metled at 250°F. (394°K.) immediately prior to being combined with the resin. The mixing ratio is 100 grams of 7343 resin to 11 grams of 7139 curing agent. The resultant clear homogeneous adhesive has an average pot life of 3 hours at 75°F. (298°K.) After brushing a 5 mil (1.27×10^{-4}M) (approximate) thick layer of the adhesive onto both surfaces, the casings were assembled and air cured at room temperature under a 2 PSI (1.38×10^{-4} newtons per M^2) compressive load for 24 hours. Optimun strength is reached after 3 days at 75°F. (298°K.) As in the previous contracts, the 2 PSI (1.38×10^{-4} newtons per M^2) loading was achieved by the use of sufficient weights positioned on a weight board such as to apply a compressive load only on the adhesive area. Foam rubber weatherstrip was used to concentrate the force at the panel joints to avoid damaging the casing materials during the pressure cure cycle.

Each SEMI panel contained an evacuation port located at the warm end. The panel seal-off device provides a one inch (2.54×10^{-2}M) diameter vacuum port, sealed with a Viron A o-ring. A steady leak rate of 4×10^{-7} atm. cc. air/second was determined for the device using a Vecco Model MS-9 helium leak detector. The seal-off device is shown in Figure 15. A demonstration model was assembled and disassembled several times to determine repeatability and leak tightness of the closure. In subsequent leak tests, the seal-off device satisfactorily demonstrated repeatability. These aluminum seal-off plates were solvent wiped with MEK and primed with a Dow Corning solution. The solution consisted of .02% by weight of Dow Corning Z 6020 resin in Ethyl Alcohol. The prime was air dryed for 45 minutes prior to the Narmco adhesive application.

Figure 19 is a photograph of one of the circumferential panels prior to evacuation. The MAM laminate is apparent in the photographs at the widest portion of the panel. Variations in panel width accomodates shingling the SEMI panel in the tank longitudinal direction in addition to shingling the panels in the circumfernetial direction. Figure 20 shows one of the polar panels prior to evacuation. The raised surfaces appearing in the photographs are thermo-vacuum formed in the casing prior to panel assembly. The reason for this as previously discussed is to provide extra casing material, thereby reducing possible residual compressive loadings on the insulation. An evacuated circumferential panel is shown in Figure 21. The mottled appearance of the panel is due to surface irregularities caused by the criss-crossed foam webs.

Inner Skirt Panels

The inner skirt panels are of a complex double curvature reverse bend shape and consequently were very troublesome to fabricate. After several futile attempts to assemble a leak tight casing enclosure, they were constructed to achieve a panel that would demonstrate mechanical and thermal performance. However, since the panels were not leak tight, the panel cryo-pumping capability was not demonstrated. The foam spacer and radiation shield layers presented very few problems. Some of the problems concerning casing fabrication were attributable to the small size. As originally designed, the inner skirt panel casings were to be fabricated on a three dimensional plywood layup fixture as shown in Figure 22. This was to be accomplished by bonding several casing segments together with Narmco adhesive. The two psi pressure required to achieve the bond line was to be obtained by clamping the mating male shape to the female mold. This method of casing assembly, although satisfactory with resepct to the thickness of most of the Narmco adhesive joint, resulted in several small adhesive filled wrinkles. These wrinkles were caused by non-uniformily forcing the two dimensional casing material into a three dimensional form. Since Narmco adhesive performance at cryogenic temperature is dependent on minimum joint thickness, the adhesive filled wrinkles were judged to be unsatisfactory for this application.

The second method to fabricate leak tight casings consisted of vacuum forming the entire casing segment from one piece of flat casing. The previously fabricated plywood form was completely filled with a foam-in-place urethane, to provide a female mold for vacuum forming. (See Figure 23) This method was found to be unacceptable because of severe wrinkles that formed, causing leaks which could not be eliminated.

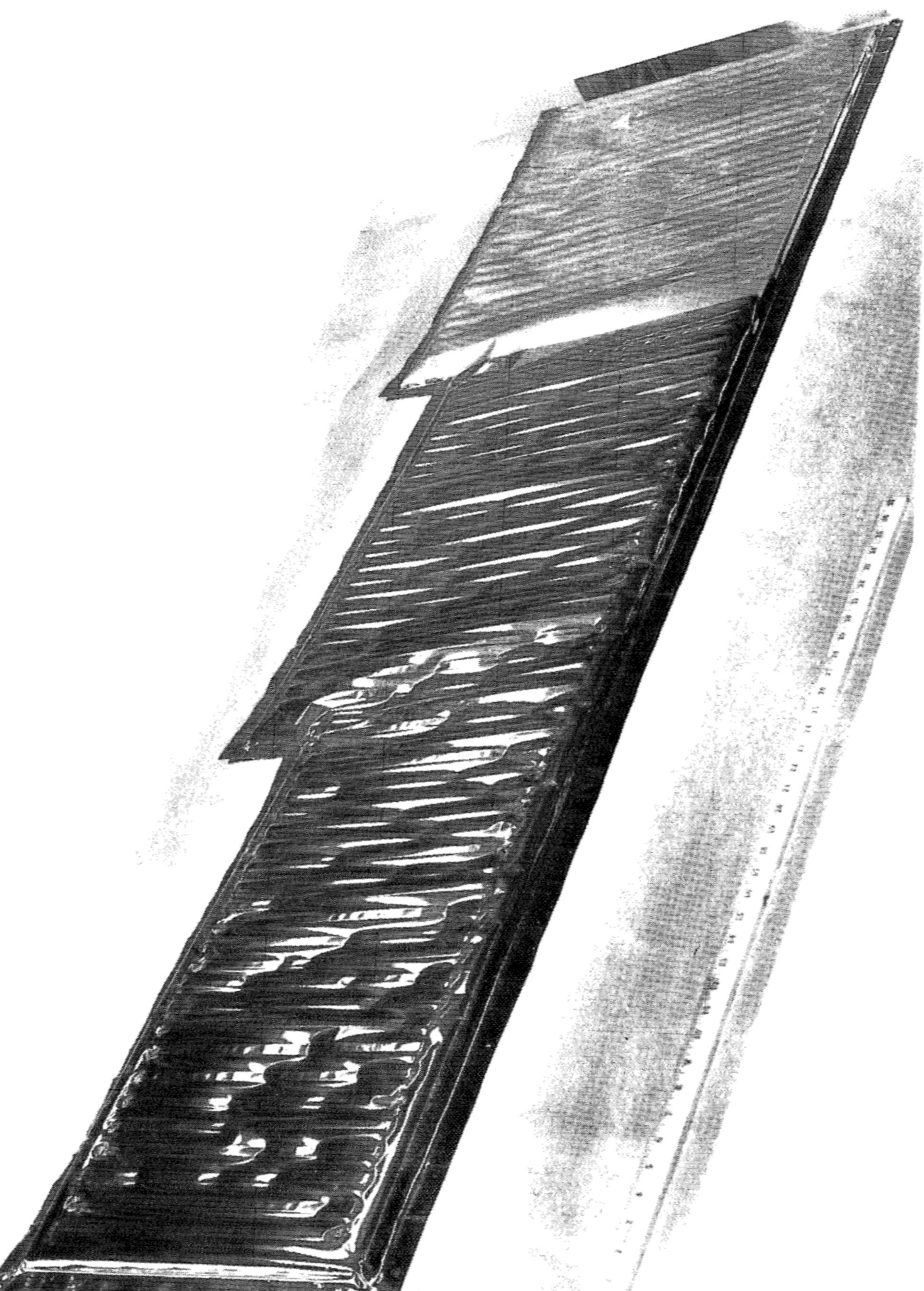


FIGURE 19. Unevacuated Circumferential SEMI Panel (217-70)

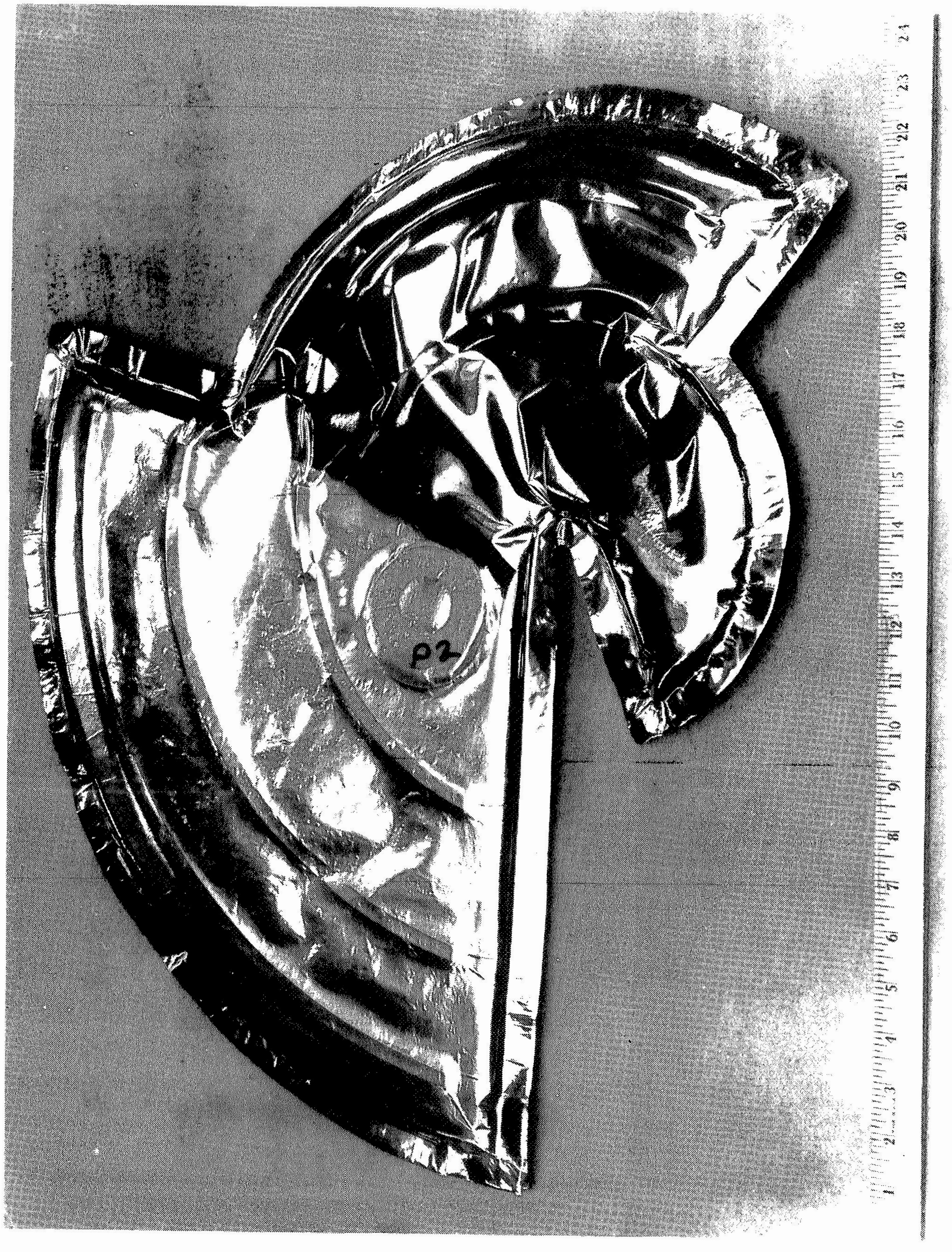


FIGURE 20. Unevacuated Polar Panel
(221-70)

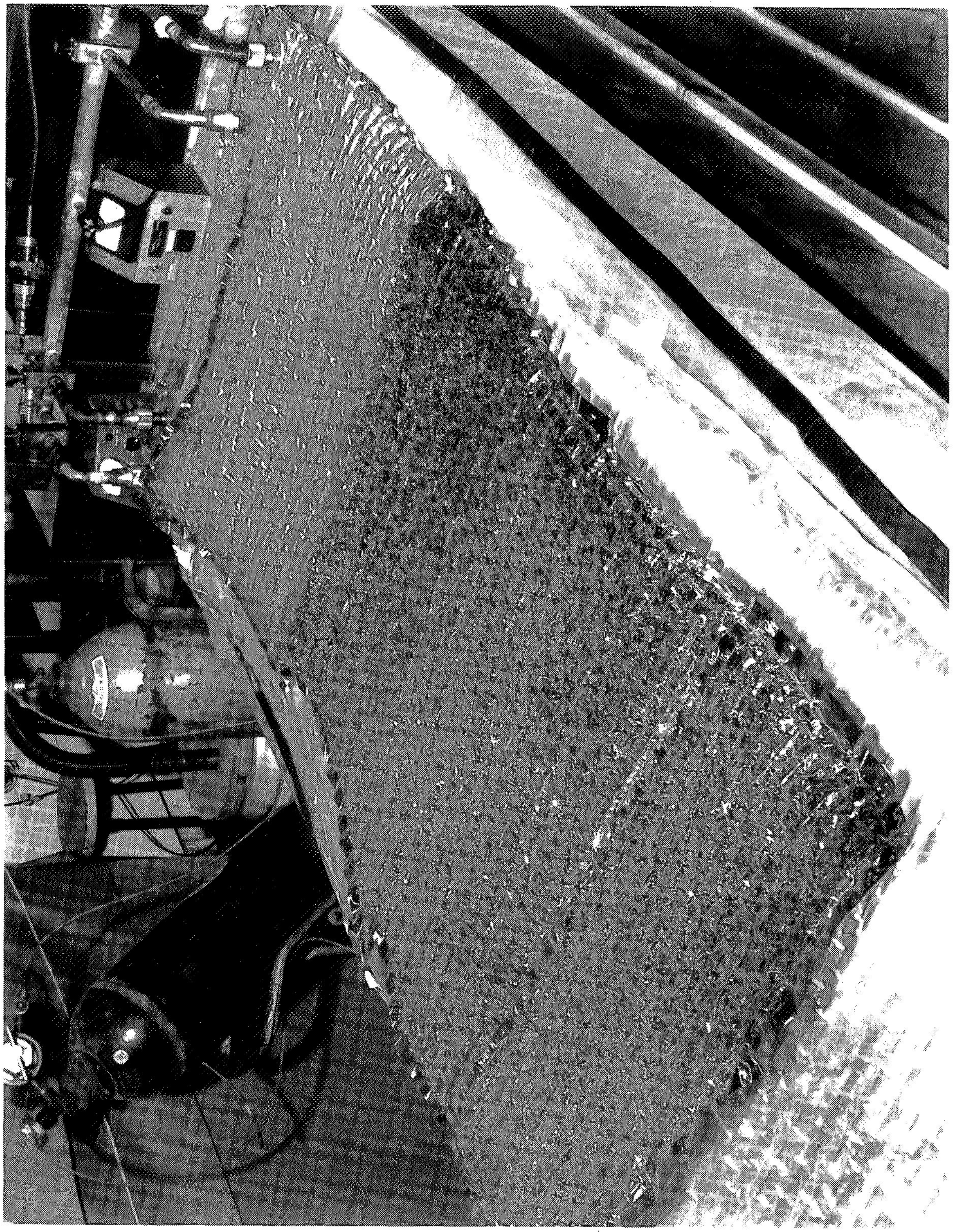


FIGURE 21. Completed Panel (Evacuated)
(219-70)

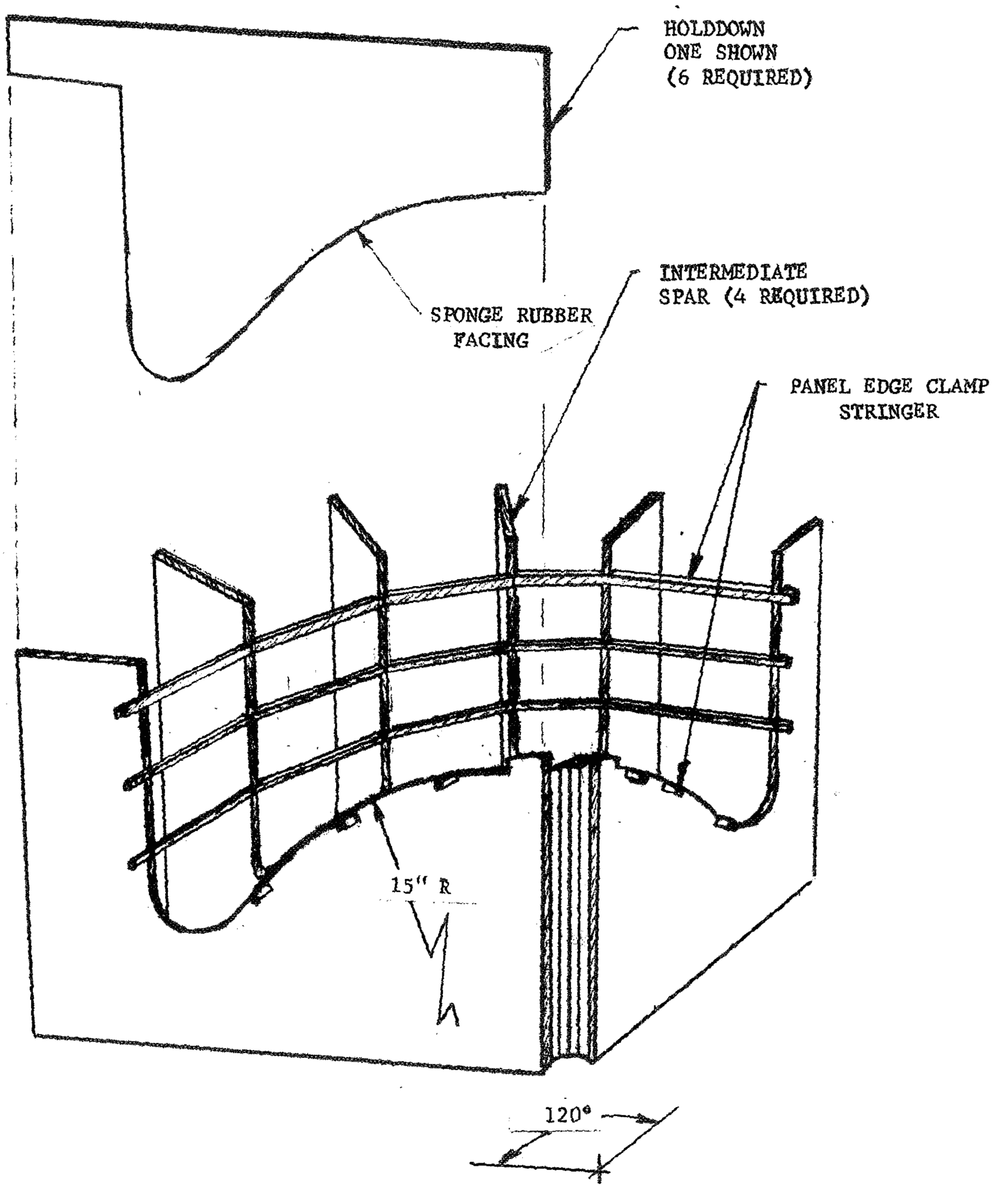


FIGURE 22. SKETCH -
FABRICATION FIXTURE
INNER SKIRT PANELS
NAS 3-12045
TASK III

Fabrication of casings for the nine inner skirt panels was finally accomplished using Goodyear 207 heat sealable adhesive joints, joining several segments into one panel casing. The resulting joints were not leak tight or even evacuable, but merely provided sufficient strength to retain the foam spacer and radiation shield layup.

Fabrication of the inner skirt insulation was accomplished using the foam plywood form as a layup mandrel. The assembly procedure followed was to lay-up the seven composite layers of foam and six radiation shields, one layer at a time. Each layer had cutouts which were overlapped to form a continuous insulation, staggering the cutouts, and overlapping the joints, such that the resulting material build-up was spread out evenly over the complete panel. After placement of all the insulation layers, the outer casing was positioned and a heat sealed adhesive joint was completed. The resulting product was an insulation blanket having the necessary number of layers, and of the proper compound curvature to insulate the inner skirt area of the model tank. Since the inner skirt panels were not evacuable, a separate impermeable barrier was installed to complete the impermeable outer layer of the air exposed inner skirt panels after the panels were installed on the Model System.

4.2.2.3 Panel Installation and Seal Off

Upon completion of the Model tank Acceptance Tests (see Section 4.2.3.2) the skirt section and also the foam in place fairing at the skirt transition was completed. A 1-1/8 inch (2.86×10^{-2}M) Vecco valve and coupling, as well as instrumentation feed throughs were added to the 30 inch (.75M) diameter stainless top sheet. The 36 gage copper constantan thermocouples and the Rosemont Engineering Temperature sensors were mounted per Figures 24 and 25. The completely insulated tank ready for shipment is shown in Figure 26.

The completed SEMI panels, i.e. circumferential and polar panels were backfilled with Coleman grade carbon dioxide to one atmosphere immediately prior to installation. Panel evacuation and backfill was completed through the panel seal off. After evacuation and Carbon Dioxide backfill, a small polyurethane bag was placed over the seal off, and purged with Carbon Dioxide. The evacuation adapter was removed and the seal off cover installed. (See Section 4.2.1.3.2). The purge bag technique allowed the evacuation port to be replaced with the seal off cover, without contaminating the backfill gas since both the panel and the bag contained carbon dioxide at one atmosphere. The inner skirt panels were not leak tight (see Section 4.2.2.2) and therefore were not backfilled. All panels were installed on the tank using VELCRO fasteners, and the panel to panel joints were sealed as discussed below.

After placing 3 inch (7.62×10^{-2}M) length of the "loop" portion of VELCRO fastener on the backside of all panels in the pattern shown on Figure 13 (Section 4.2.1.3.2) and placing 3 inch (7.62×10^{-2}M) lengths of "pile" in a matching pattern on the tank, the 8 panels were sequentially wrapped one third of the way around the tank in a clockwise direction as viewed from the flange end. In subsequent stages, the "pile" was placed on the previously installed "loop" portion (panel backside), the adhesive was activated with Methyl Ethyl Ketone (MEK) and the panel wrapping was continued. In this manner, mating of the VELCRO fasteners was assured since proper alignment was established at

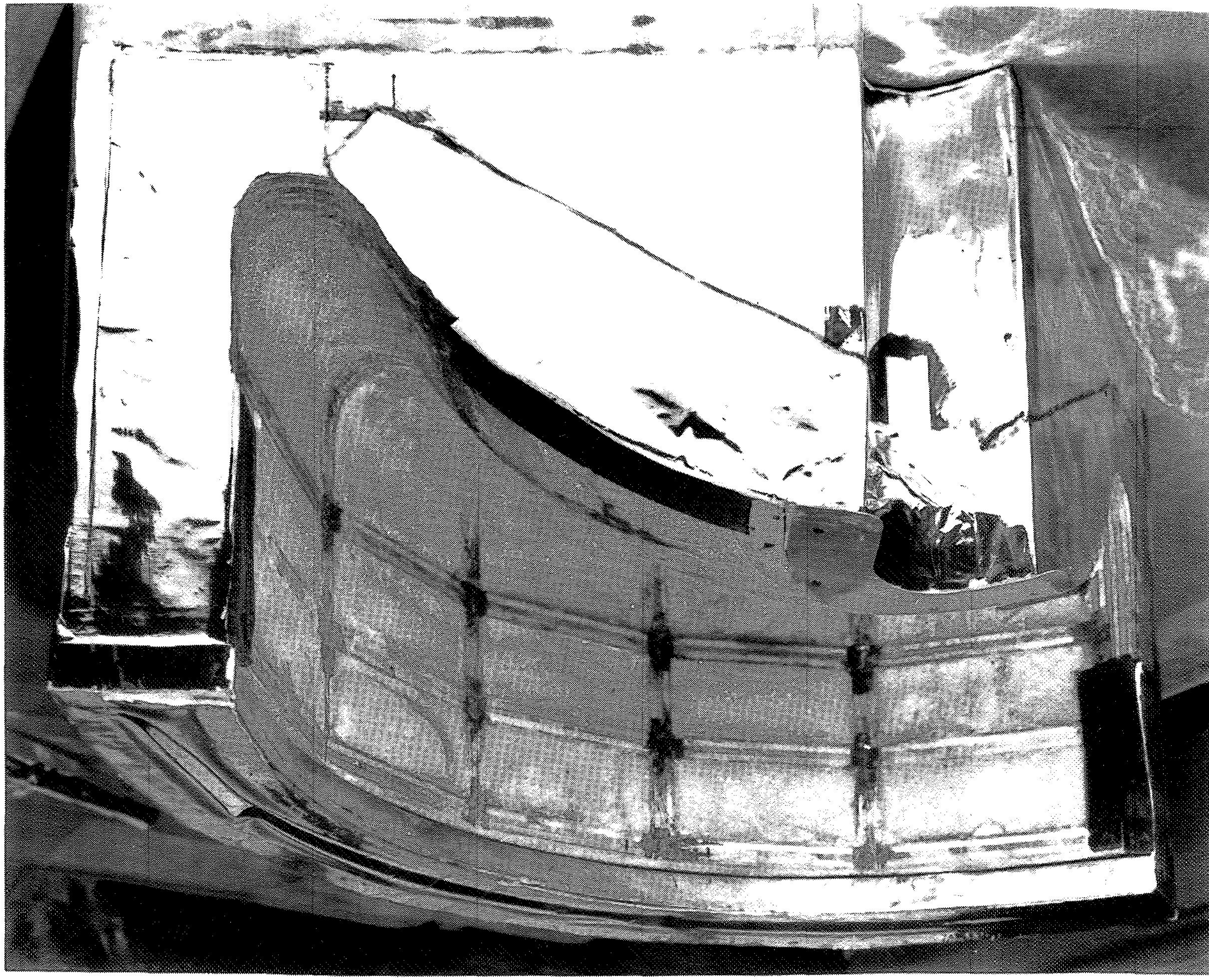


FIGURE 23. Inner Skirt Casing Vacuum Fixture (220-70)

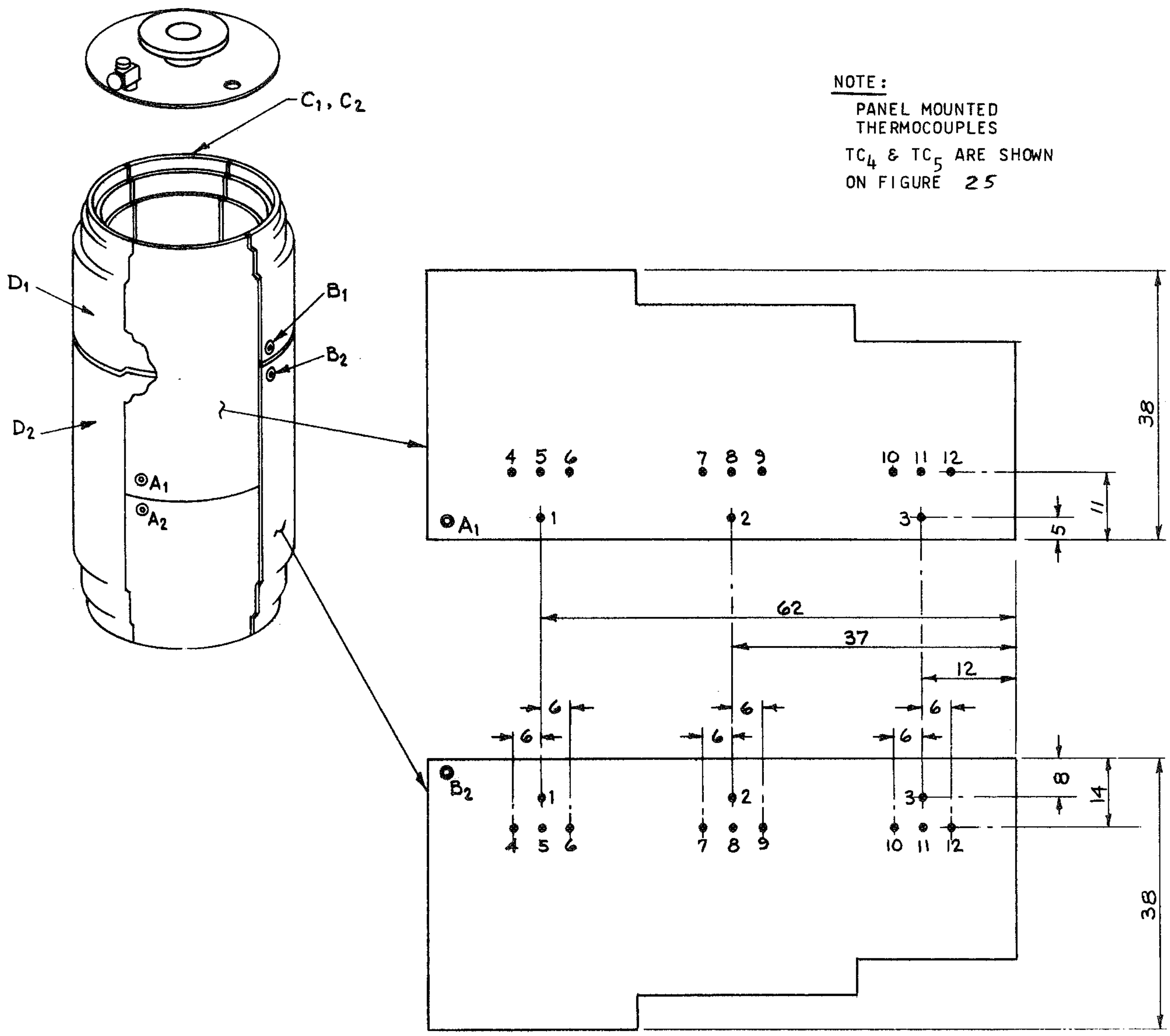
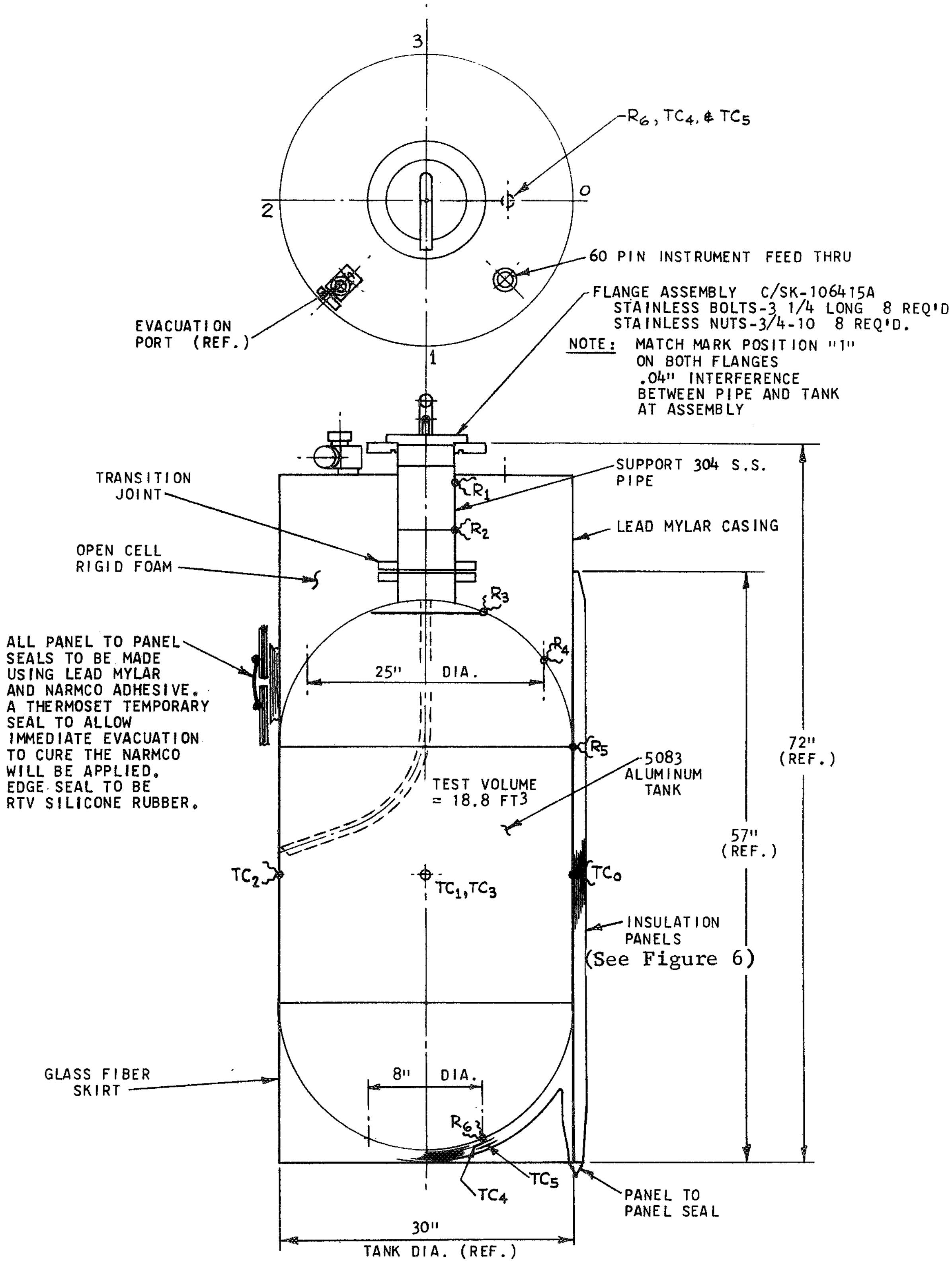


FIGURE 24 CIRCUMFERENTIAL
PANEL THERMOCOUPLE POSITIONS
TOTAL 24 THERMOCOUPLES



NOTE: TC_4 & TC_5 THERMOCOUPLES ARE PANEL MOUNTED.

FIGURE 25. MODEL TANK ASSEMBLY DRAWING TEST TANK THERMOCOUPLES & ROSEMONT TEMPERATURE SENSOR POSITIONS (EXTERNALLY MOUNTED)

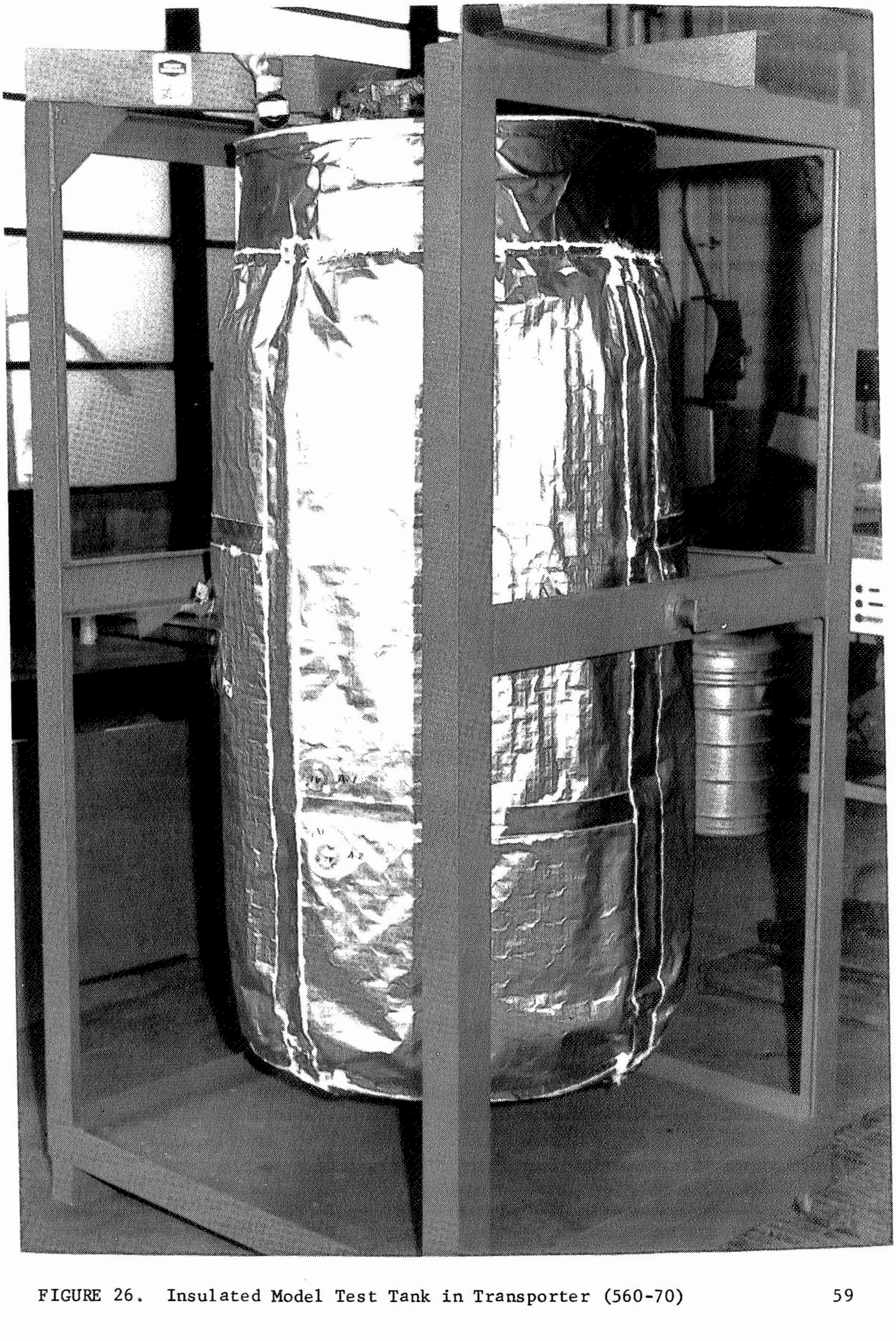


FIGURE 26. Insulated Model Test Tank in Transporter (560-70)

assembly. All panels were removed after the initial placement, to inspect the bonding of the VELCRO to the panels and tank. In several closures, inspection revealed incomplete bonding of the VELCRO to substrate. These bonds were again activated with MEK and allowed to cure. The panels were then re-installed in the same manner i.e. sequentially wrapping the panels around the tank.

The nine inner skirt panels were installed using VELCRO fasteners to attach the panels to the tank and to each other. VELCRO placement was as shown in Figure 14 (Section 4.2.1.3.2). Assembly procedure was similar to that used in placing the circumferential panels.

The three polar panels backfilled with Coleman grade carbon dioxide to a one atmospheric pressure, were installed using VELCRO fasteners, placed as shown in Figure 14 (Section 4.2.1.3.2). The panel geometry required that the three panels be assembled to each other, and then installed on the tank as a unit.

Panel to panel seals were completed as described on Model Tank Assembly drawing (Figure 25). One exception was the inner skirt panels, which because of fabrication difficulties encountered were not evacuable nor laminated with a Mylar aluminum foil Mylar surface sheet over the standard 4 ply aluminized Mylar laminate. Rather than the MAM laminate, a Mylar-lead-Mylar (MLM) laminate was installed separately after the inner skirt panels had been installed. The MLM was sealed to the polar panels and the circumferential panels thus completely sealing off the dummy support skirt interface.

4.2.3 Test

4.2.3.1 Panel Leak Tests

In order for the Self Evacuating Multilayer Insulation (SEMI) system concept to be operable, the cryopumping gas must remain (1) within the system and (2) relatively free from external contamination. Upon examination, both of these items require the use of a leak tight casing, with the degree of leak tightness dependent upon the expected life, and/or allowable heat leak and weight penalties. During work on the previous SEMI system contracts, a casing system concept was devised, whereby a 4 ply laminate of aluminized Mylar is used for the entire panel, and the air exposed portion of the casing, (outer 1/3 of the panel) is further protected against air permeation by a second laminate of aluminum foil and Mylar. The second laminate of aluminum foil and Mylar is laminated to the 4 ply casing material using the Goodyear G 207 adhesive (see Section 4.2.2.3).

Each circumferential and head panel was helium leak checked. Because of the relatively high helium permeability of the 4-ply casing material, (3.5×10^{-6} atm. cc. helium/sec.-ft^2) as compared to the MAM (2×10^{-9} atc. cm^3 helium/sec.-ft^2 atm.), a mixture of nitrogen gas containing slightly less than 1% by volume of helium gas was used for helium leak checking. By using this method, a helium leak rate was established for a relatively permeable material, by a ratio or the measured indicated leak rate to the concentration of helium in the trace gas. For the SEMI panels, the panel leak rate compared favorably with the established leak rate of the parent 4-ply casing material. Table 5 lists the panel total leak rate. This leak rate includes casing permeability, joints and leakage through the evacuation seal-off o-ring. The helium leak rate for the parent casing materials is 0.3 to 0.5 x 10^{-5} atm. cc. helium/sec. ft^2 atm. helium.

4.2.3.2 Tank and Related Hardware Tests

The Model System demonstration tank was subjected to helium leak tests, as well as hydropressure tests, static dead load tests, and vibration tests as defined in the Tank Acceptance Test Plan A/SK 106462. (See Appendix 6) All testing was completed as presented in the test plan.

Helium Leak Test

The tank was evacuated for approximately 72 hours prior to the helium leak check. A small leak in the upper cylinder to head weld was located and repaired by removing metal and rewelding. After repair the leak rate was determined to be 1 x 10^{-10} atm. cc. air/sec. (1 x 10^{-7} atm. cc. air/sec. or less is acceptable.) The aluminum gasket bolted flange connection was not included in this test. The model test tank with the temporary evacuation port attached is shown in Figure 15 (Section 4.2.2.1).

After the tank acceptance vibration test, a tank leak rate of 2.3 x 10^{-9} atm. cc. air/sec. was determined. This compared to 1 x 10^{-10} atm. cc. air/sec. before vibration testing. The slight difference is probably attributible to leakage across the soft aluminum gasket between the flange assembly and the tank. In any case, the leak rate is less than 1 x 10^{-7} atm. cc. air per sec. established for the vessel and therefore the vessel was satisfactory.

Proof Pressure Test

The model tank as shown in Figure 27 mated to the vibration adapter, was filled with water and pressurized to 115 psig (7.9 x 10^{-5} newtons per M^2) for 15 minutes. Stress levels were less than 1500 psi as no change was noted in the Stress Coat, applied as per vendor instructions. (The craze point for this particular stress coat is 150 micro inches per inch, which for aluminum will indicate a stress level of 1500 psi.)

Axial Load Tests

Compression Test

With the tank positioned as shown in Figure 27, additional weights were placed on the skirt head to increase the compressive load on the support tube (as per specification A/SK 106462). The stress-coat indicated acceptable compressive stresses. Actual stresses were calculated at 2200 psi (1.515 x 10^7 newtons per M^2).

TABLE 5

Measured Total Leak Rate of Model Tank SEMI Panels

Includes: Casing Permeability, O-ring Seal-Off and Joint Leakage

Panel No.*	Size Inches	Meters	Total Leak Rate (ATM cc. Air)/sec.
A_1	38 x 75	.97 x 1.78	3.46×10^{-5}
B_2	38 x 75	.97 x 1.78	3.87×10^{-5}
C_1	38 x 75	.97 x 1.78	4.07×10^{-5}
D_2	38 x 75	.97 x 1.78	2.66×10^{-5}
A_2	19 x 75	.49 x 1.78	2.19×10^{-5}
B_1	19 x 75	.49 x 1.78	2.31×10^{-5}
C_2	19 x 75	.49 x 1.78	2.11×10^{-5}
D_1	19 x 75	.49 x 1.78	2.17×10^{-5}
P_1	Polar		3.8×10^{-6}
P_2	Polar		4.44×10^{-6}
P_3	Polar		4.54×10^{-6}

* Panel Numbers refer to Figure 24.

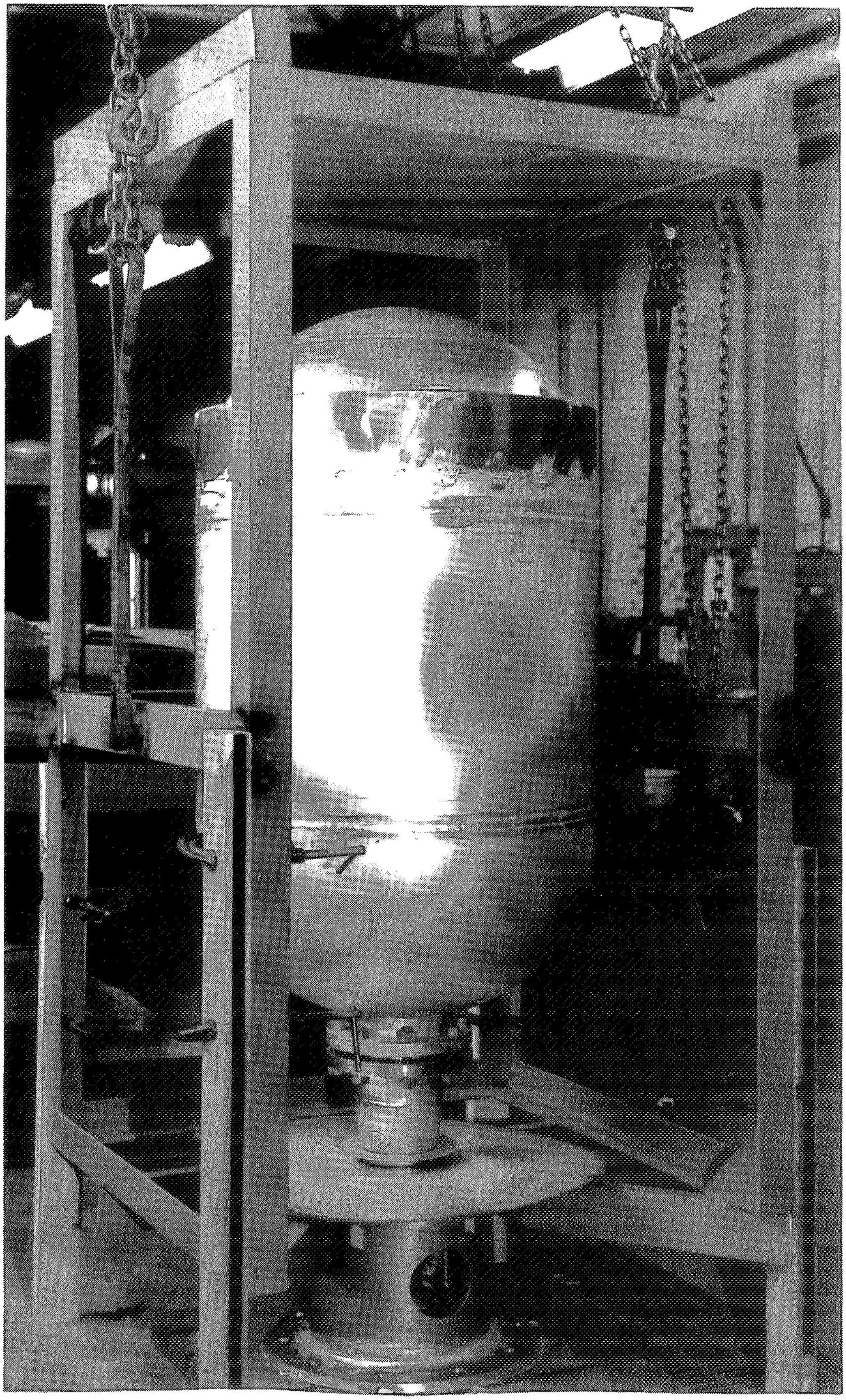


FIGURE 27. Model Test Tank Flange Down Position (247-70)

Tension Test

With the tank mounted in the flange up position, the appropriate amount of water (as noted in Test Plan) was added to increase the support tube loading in tension. Stress levels were again noted to be less than 1500 psi from observations of the Stress Coat.

Cantilever Test

To perform the cantilever test, the tank and vibration adapter sections were bolted to a weld positioner as shown in Figure 28. For this test the tank was removed from the transporter, and the transporter was then lowered to the floor. (The tank is not being supported at the mid-point.) Definite strains were indicated in the Stress Coat during this test. The stresses were lower than previously calculated, and this is probably attributable to manufacturing, rather than deficiencies in observing the Stress Coat. The junction of the support tube to the head transition piece was expected to be the highest stress zone at 19000 psi (1.31×10^8 newtons per M^2). However, due to manufacturing variations, i.e., a larger fillet at the weld area than specified, the maximum measured strain indicated by the Stress Coat was 1100 micro-inches (2.79×10^{-5} M) or 11000 psi ($7.5 \times 10^{+7}$ newtons per M^2) for a given side load of 5g in bending. The strain level at the weld junction of the support head to the head transition joint was expected to be 1700 micro-inches (4.32×10^{-5}M) however, this was measured to be 700 micro-inches (1.77×10^{-5}M) or 7000 psi (4.81×10^{-7} newtons per M^2). An attempt to measure tank deflection under load proved impossible because the weld positioner/end support member was not perfectly rigid, therefore the tank rotated about the cantilever support. This rotation was observed using two bubble levels, one located on the tan surface and the second on the face of the positioner.

Vibration Tests

The bare model test tank (see Figure 29) was subjected to vibration levels as dictated in the Acceptance Test Plan. Testing was performed by Dayton T. Brown Co., Long Island, New York. Both the 1g and 10g acceleration levels were completed and the tank was found to be acceptable with the following comments. Observation of the 1g data obtained from the two triaxial accelerometers, indicated that at times the surface loading at the tank was greater than the input loading. This amplification, apparently caused by the support head, appeared at a resonant frequency of approximately 70-80 cps. Therefore, the 10g test was conducted using the tank surface pick-up to limit the input signal to the flange. Recorded upsweep data is presented as Figure 30 and 31. Referring to Figure 30, observe that at 120 cps., for a 1g input fore and aft, the tank surface was experiencing approximately a 3g acceleration. Likewise observe that at 70 to 80 cps. the tank vertical axis was likely limiting the fore and aft input, while between 80 and 90 cps. the tank tranverse axis was limiting the input. Similar observations at a 10g level can be made from Figure 31. The complete Dayton T. Brown test report is included as Appendix 7.

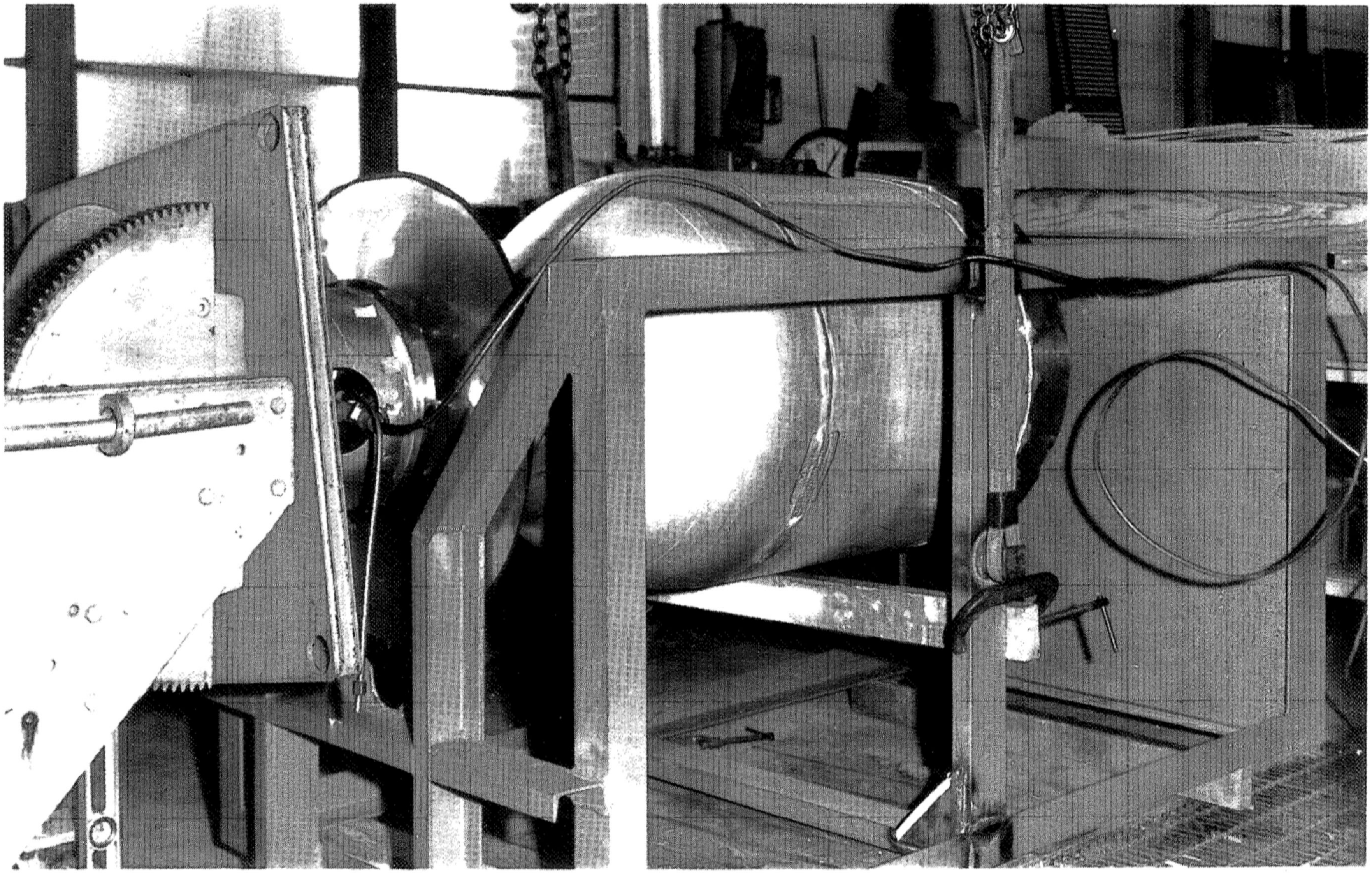


FIGURE 28. Cantilever Test-Model Test Tank

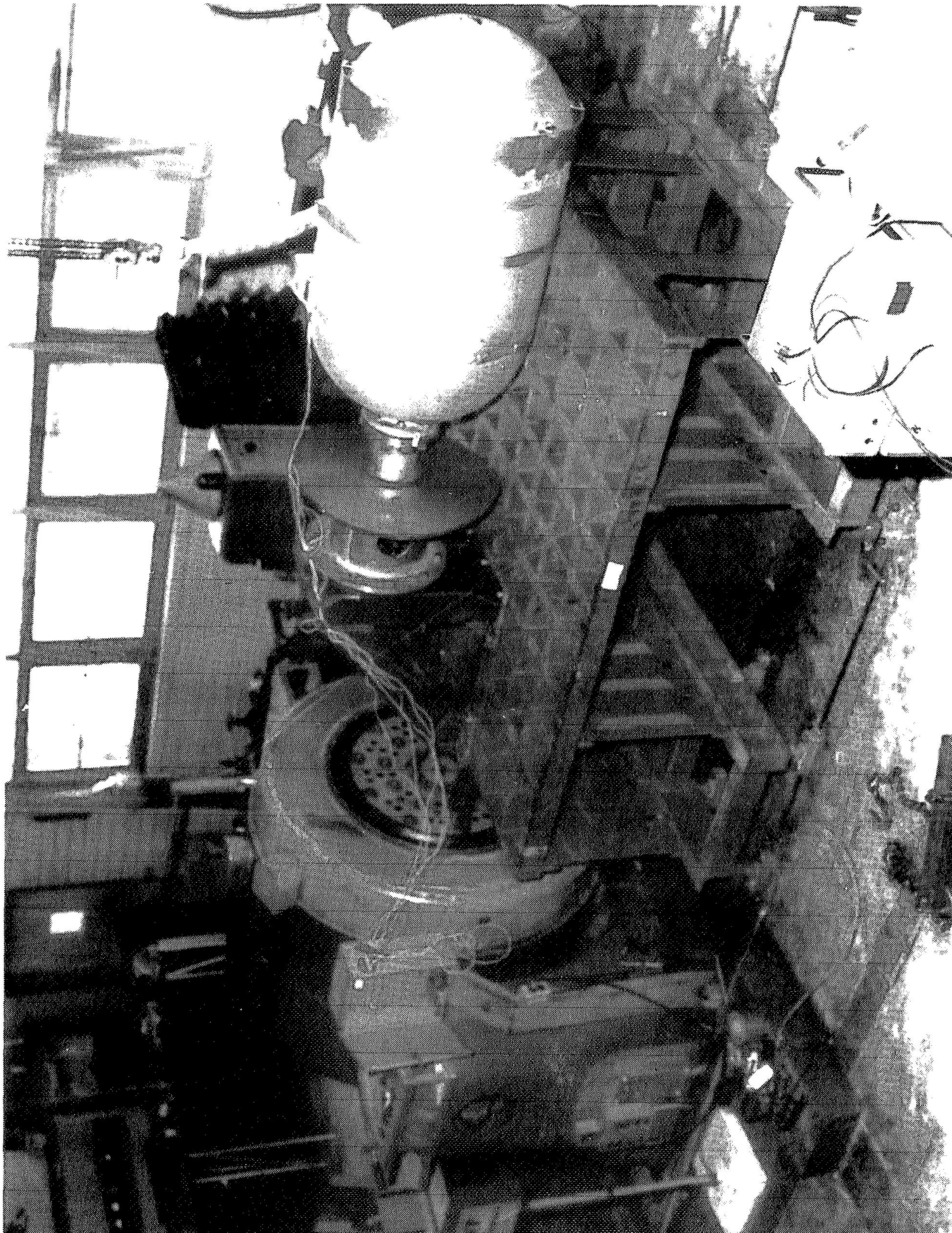


FIGURE 29. Model Tank During Vibration Acceptance Testing

DAYTON T. BROWN, INC. - TESTING LABORATORIES
TESTED FOR: UNION CARBIDE CORPORATION ITEM: TANK
MFG: UNION CARBIDE CORPORATION MODEL: TEST S/N: 629
TYPICAL VIBRATION TEST SETUP; TEST UNIT SHOWN SECURED
TO VIBRATION EXCITER IN THE FORE & AFT TEST PLANE
DTB JOB NO.: 8565 PHOTO FILE #: 4287 DATE: 15 APRIL 1970
DTB03R70-0587 ENCLOSURE: 3 PHOTOGRAPH 1 of 1

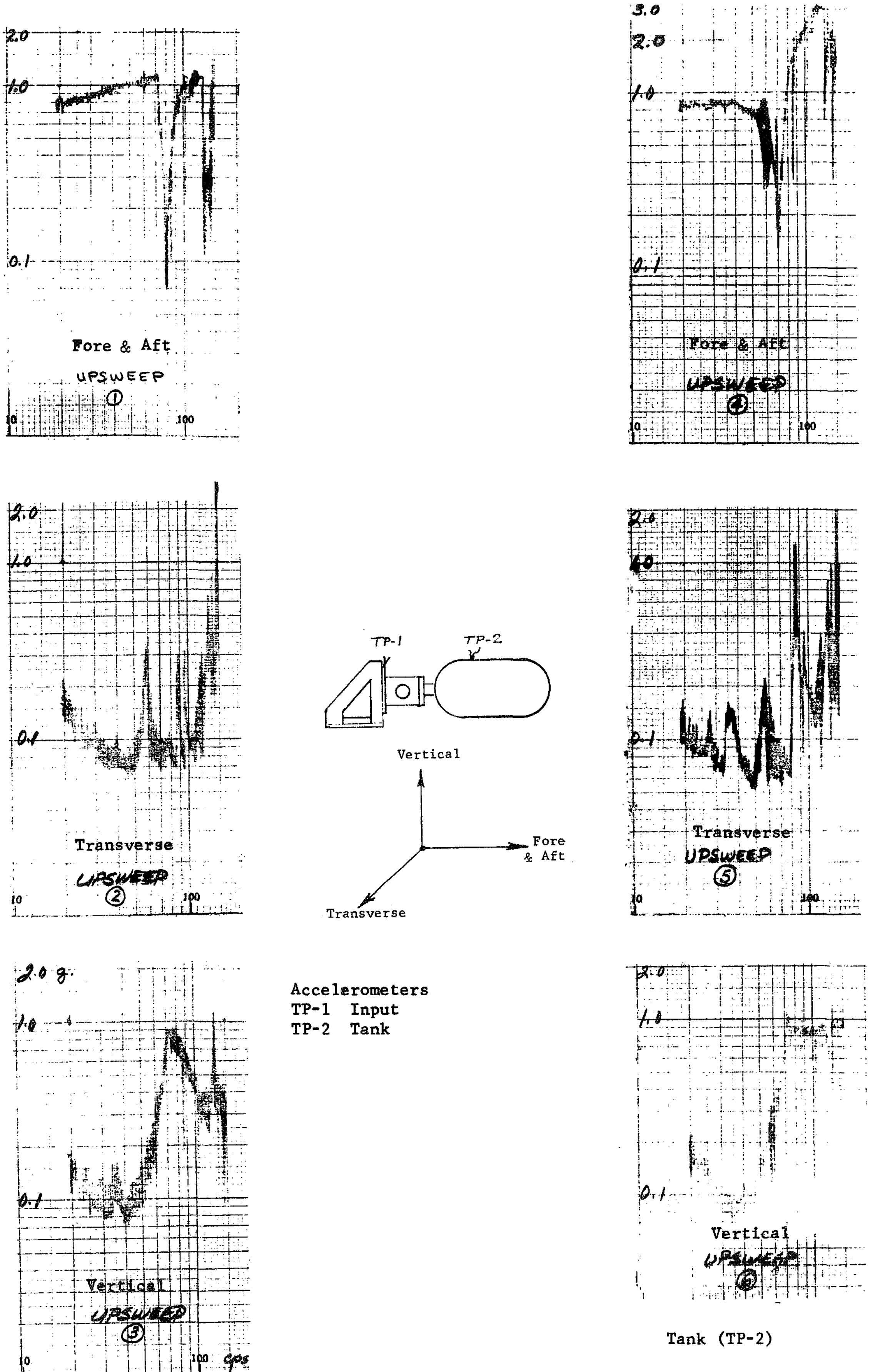
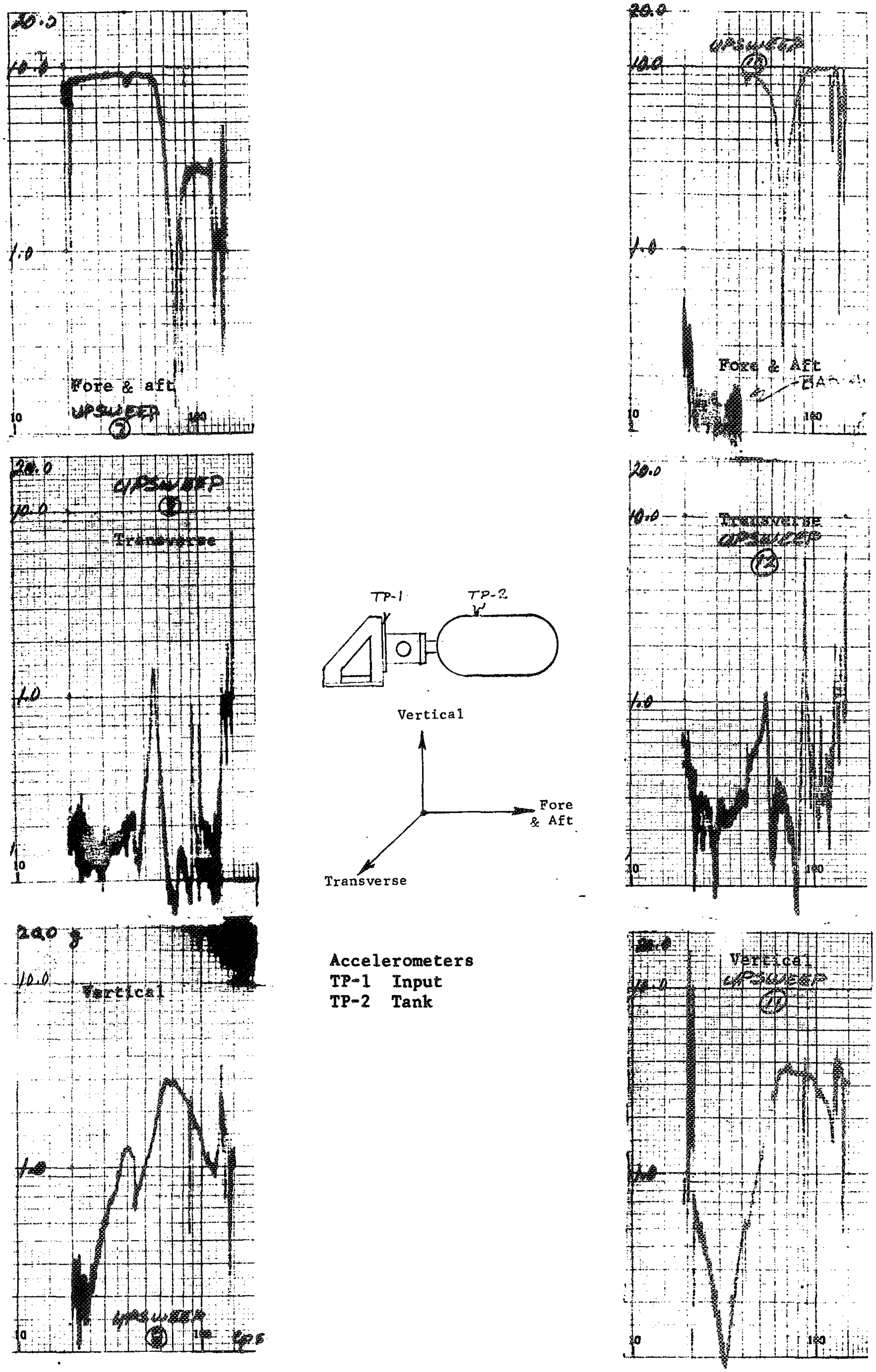


FIGURE 30. Vibration Upsweep Trace - 1g Level
① Refers to graph page Dayton T. Brown Report



Input (TP-1)

Tank (TP-2)

FIGURE 31. Vibration Upsweep Trace
10g Level
Refers to graph page Dayton T. Brown Report

4.2.3.3 Installed SEMI System Evaluation Tests

Testing of the Model System involved two NASA test sites, namely NASA Plumbrook Station "K" site, Sandusky, Ohio and the Goddard Space Flight Center, Launch Phase Simulator Facility, Greenbelt, Maryland. Evaluation consisted of the initial thermal test, the Launch Profile test, and a final thermal test. The purpose of the thermal testing was to evaluate degradation of the SEMI system resulting from the simulated launch environment. Description of tests performed and the results are included in the following sections. The Launch Phase Simulator Test Plan and the Thermal test plan are included as an appendix to this report.

4.2.3.3.1 SEMI System Thermal Test #1

The model system was thermal tested in the "as received" condition immediately prior to dynamic testing. (The insulated tank was stored at NASA's Plumbrook Station in a CO_2 purge bag for approximately 5 months after fabrication.)

The system was tested by NASA personnel as described in Test Plan YOR 1327 (See Appendix 8). Briefly, the tank was mated to the A.D.L. calorimeter cold guard flange, and a helium purge bag installed to enclose the interface flange and cold guard. The purpose of the helium filled cold guard purge bag was to prevent nitrogen gas in the Plumbrook chamber from condensing on the LH_2 cold guard during the initial cooldown. The chamber was evacuated one hour after filling the measure tank (test item) with LH_2. The guard tank was not filled with LH_2 until 2 hours after the chamber was evacuated. The space behind the panels developed leaks, and was not continuously evacuated during the test, and probably accounts for the overpressure behind the panels resulting in blowing out the polar panels on warm-up. The model system after thermal test, with cold guard purge bag installed is shown in Figure 32.

The following readings were obtained approximately 24 & 48 hours after the start of the test:

	24 Hours	48 Hours
Chamber pressure	3×10^{-6} torr	4.5×10^{-6} torr
Insulation Pressure		
Panel A-1	7.8×10^{-4} torr	4×10^{-4} torr
Panel A-2	3.7×10^{-4} torr	5.8×10^{-4} torr
Measure Tank Boil Off	52 Btu/Hr. (15.25 Watts)	45 Btu/Hr. (13.2 Watts)

Using the figure of 45 Btu per hr. (13.2 Watts), the calculated average panel heat flux as determined by this test was 1.3 Btu per hr.-ft^2 (4.1 Watts per M^2). The computer predicted average panel performance was 1.5 Btu per hr.-ft^2 (4.73 Watts per M^2). It is evident that measured thermal performance exceeded the

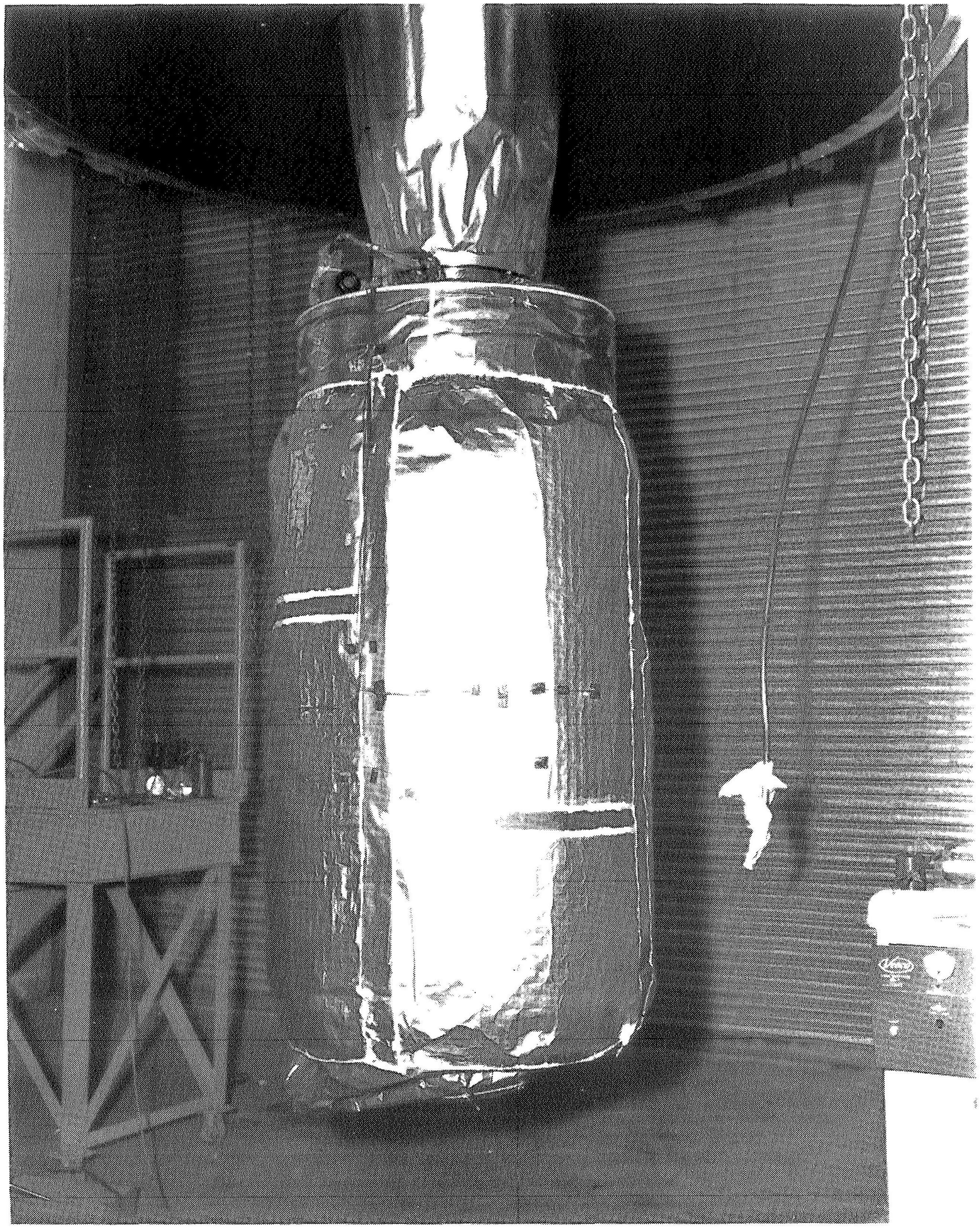


FIGURE 32. Model System after Thermal Test - Cold Guard Purge Bag Installed. (P70-2176)

computer calculated performance, however, the reason for this is not immediately known. A thermal shroud including radiation shields was placed around the insulation and could have influenced the results significantly. In any case, the same set-up was used for the thermal test after the dynamic tests were conducted at Goddard.

4.2.3.3.2 SEMI System Dynamic Tests

The model system was subjected to dynamic tests at NASA's Goddard Space Flight Center. The SEMI system was vibration tested in a so called "off board" vibration only test mode (see Figure 33), and also tested in the Launch Phase Simulator Chamber where the system was subjected to a combined environment of acceleration, vibration, acoustics and a launch evacuation profile. Testing in both modes was performed with the tank cooled to liquid nitrogen temperature. The SEMI system is shown in Figure 34 mounted to the movable end cap of the test chamber, being readied for mating with the chamber.

The SEMI system was tested per Test Plan DIRS 02198 (see Appendix 9). Actual testing deviated only slightly from this plan, although, accidently the unit experienced a 10g loading in all three mutually perpendicular axis during the off board test mode. The lower head insulation was observed to have stayed in position during and after the off board vibration tests; it did not pop out upon warmup as was noted after the first thermal test at Plumbrook. However, before startup of the on-board vibration test in the Launch Phase Simulator, the lower head was observed to have popped after the LN_2 cooldown. Both vibration tests were conducted with the test tank pre-cooled with liquid nitrogen. The off board tests were of a short duration, and, as indicated by a study of the temperature profile, the tank was cooled only to liquid oxygen temperature, and for only a short period. This probably allowed very little gas to condense on the cold surface, resulting in insufficient gas pressure in the space behind the panels to pop the panels upon warm up. However during the on-board vibration test, because of the complexity of the set up, the tank was cooled and maintained at liquid nitrogen temperature for several hours prior to start of the dynamic test. During this period, the system was exposed to ambient air and probably resulted in condensing a large quantity of air on LN_2 surfaces. The condensed air was then quickly vaporized when the LN_2 was withdrawn from the tank at the start of the dynamic test, resulting in an increase in pressure in the space behind the panels, which subsequently popped the panels. Viewing the test movies, it was noted that prior to tests the inner skirt and polar panels were stretched very tight, and these panels were convex rather than concave as originally installed. This overpressure resulted in placing additional forces on the circumferential panels, and probably contributed to the failure of panels C_2 and D_2, at Goddard.

It is interesting to note that the SEMI System endured the extreme physical testing even with the polar panels reversed (due to the overpressure) and consequently the panels were completely unsupported except for the inner support skirt casing attachment around the circumference of the tank. Except for several small scraps of insulation that were observed in the Goddard test movies, (later determined to be from panel C_2) the insulation system appeared to be intact, with no apparent damage to panels or panel-to-panel seals. (See Post Test Evaluation, Section 4.2.4).

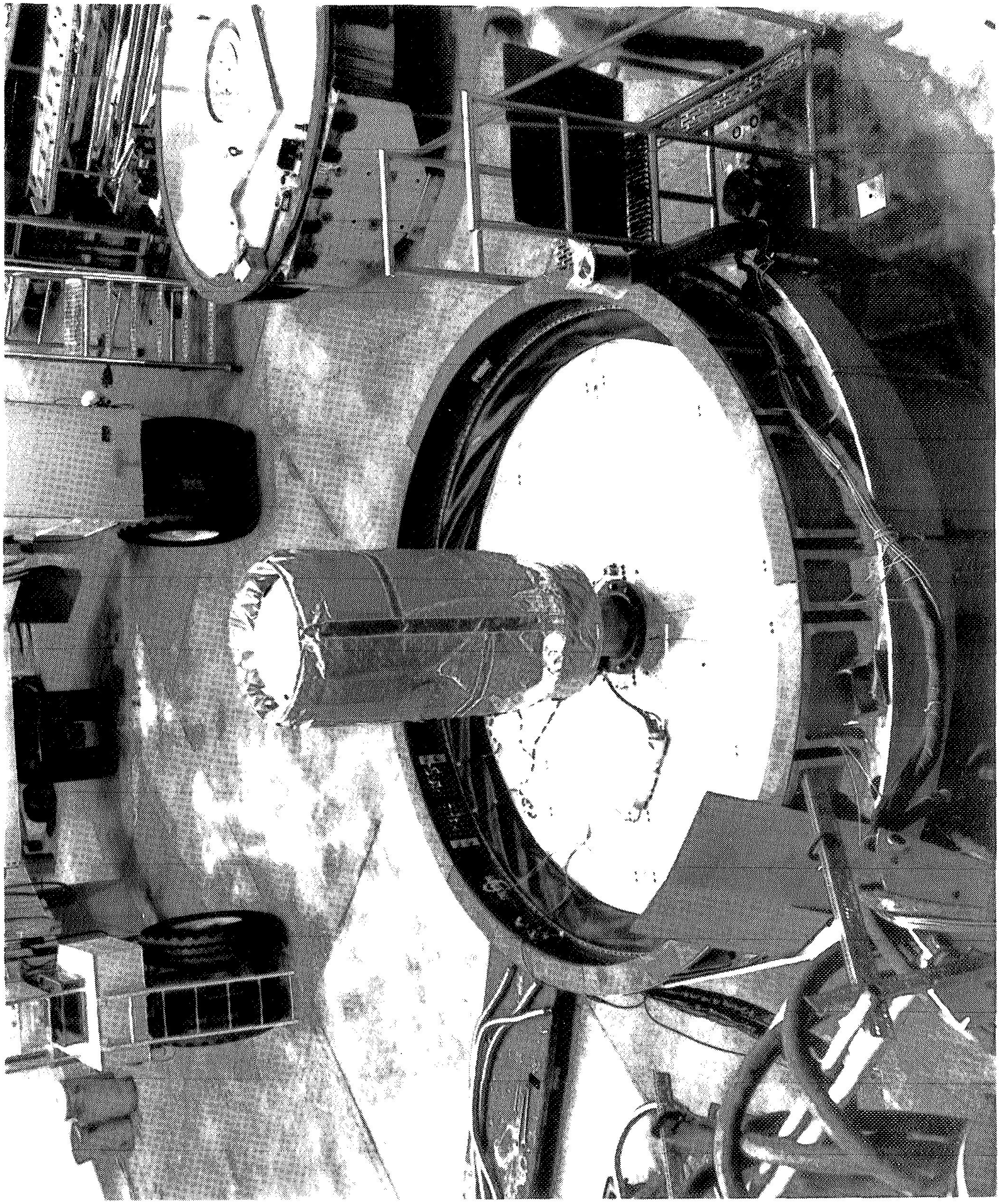


FIGURE 33. SEMI System "Off Board" Vibration Test Mode.
(G 71-2382)

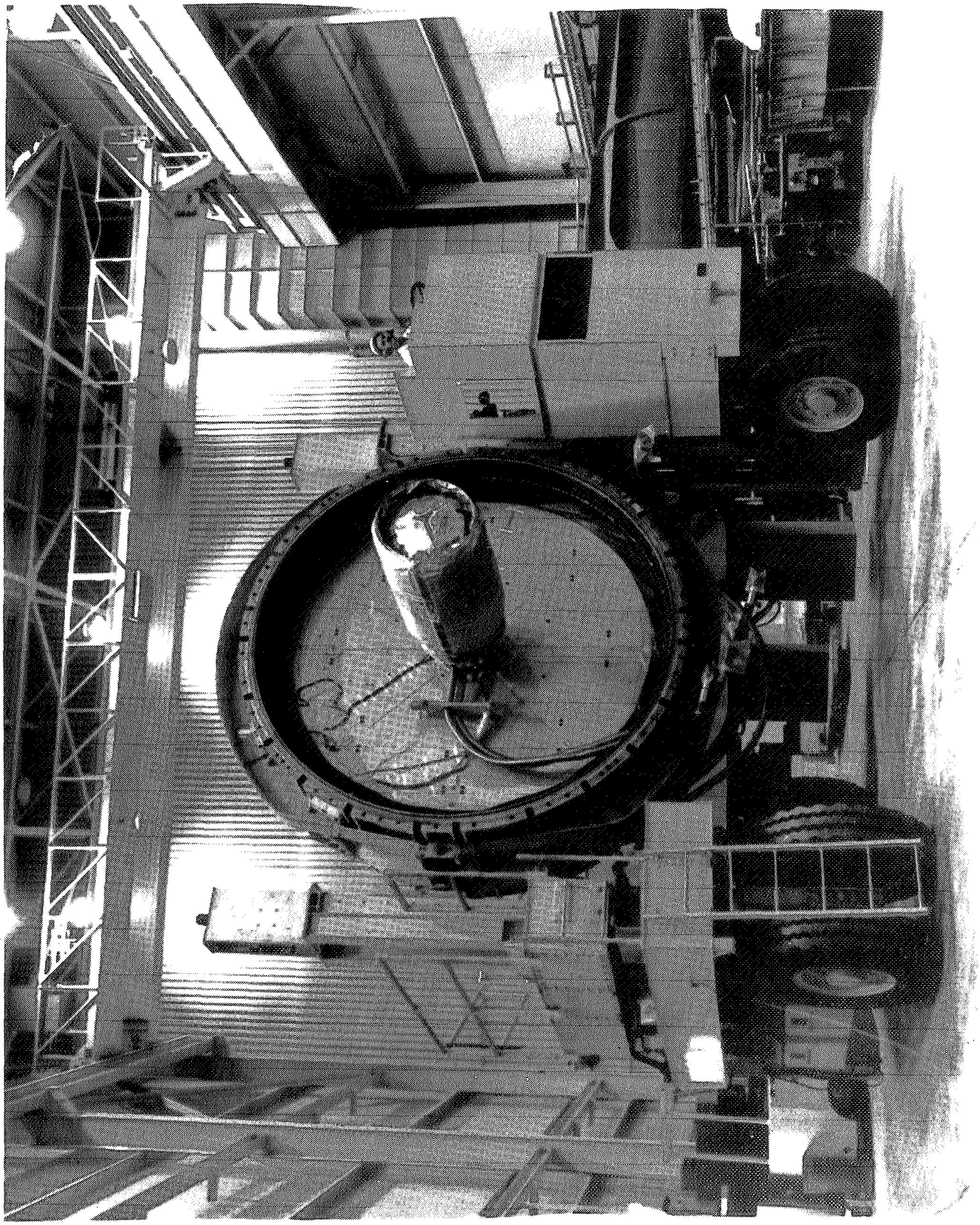


FIGURE 34. SEMI System Mounted on End Cap Ready to Mate With the Goddard Test Chamber. (G71-2379)

4.2.3.3.3 SEMI System Thermal Test #2

Upon completion of the dynamic tests, the SEMI system was returned to NASA Plumbrook's K site and re-installed in the test chamber. In view of the fact that several of the SEMI panels had been torn it was decided to deliberately cut a hole in the casing of all panels to assure a low insulation vacuum, and to allow the chamber to be evacuated prior to loading liquid hydrogen. The test was run with the chamber at 6.9×10^{-6} torr, panel A-1 was reading 2.4×10^{-4} torr, (A-2 was not operating). After 48 hours, the chamber pressure had reached 1.6×10^{-6} torr, A-1 panel was reading 5.4×10^{-5} torr, and the boil off was 42 Btu per hr. (12.3 Watts). This performance compares to 45 Btu per hr. (13.2 Watts) achieved prior to the dynamic tests. Apparently, the insulation improved slightly as a result of the vibration testing. This slight improvement is probably traceable to a reduction in solid conductivity through the insulation, or perhaps to a more fully recovered insulation system.

The overall system thermal performance was again better than calculated using the computer program (see Section 4.2.1.2). Calculated total heat gain for the system was 66.3 Btu per hr. (19.4 Watts) as opposed to a measured steady state boil off of 42 Btu per hr. (12.3 Watts) for this test. As with the previous test, a study of the data plots does not reveal the reason for the improved thermal performance, as all thermocouple and Rosemont temperature sensor readings appear to be in order. The average panel thermal performance for this system is calculated as 1.2 Btu per hr.-ft^2 (3.8 Watts per M^2).

4.2.4 Post Test Evaluation

Upon completion of all tests, the SEMI system was examined for damage. Very little damage was observed, with the exception that panel "C_2" had obviously incurred damage along a panel seam (foam & shield were visible) and that the polar panels, although still attached to each other, were no longer attached to the tank except along the edge seal with the inner skirt and circumferential panels. Prior to removing any panels, all of the circumferential panels were individually evacuated with a small roughing pump to evaluate vacuum integrity. After making temporary repairs on several intentionally damaged areas, all panels except C_2 and D_2 were found to be evacuable to $\sim$28 inches of mercury (9.45×10^4 newtons per M^2) within a few minutes of pumping. This was considered indicative of panel tightness, since even a small tear or cut in the casing would have had an observable affect on the achieved pressure in regards to pumpdown time and pressure obtained. Further leak tightness evaluation of the panels was determined to be impossible with the panels installed on the tank. (Although the panels were carefully removed, several were damaged slightly, including some delamination at the edge. The delamination resulted from the edge seal procedures. Consequently, the panels were not helium leak checked after removal, since a valid test could not be performed.)

The circumferential panels were removed from the tank by cutting apart one of the vertical panel to panel seal strips between the panels, and removing all eight panels as an assembly. This method resulted in minimizing the panel damage during disassembly. Panels "C_2" and "D_2" were both observed to have suffered damage in the casing joint adjacent to the fiber glass skirt extension, and that the damaged zone of both panels occurred at the same

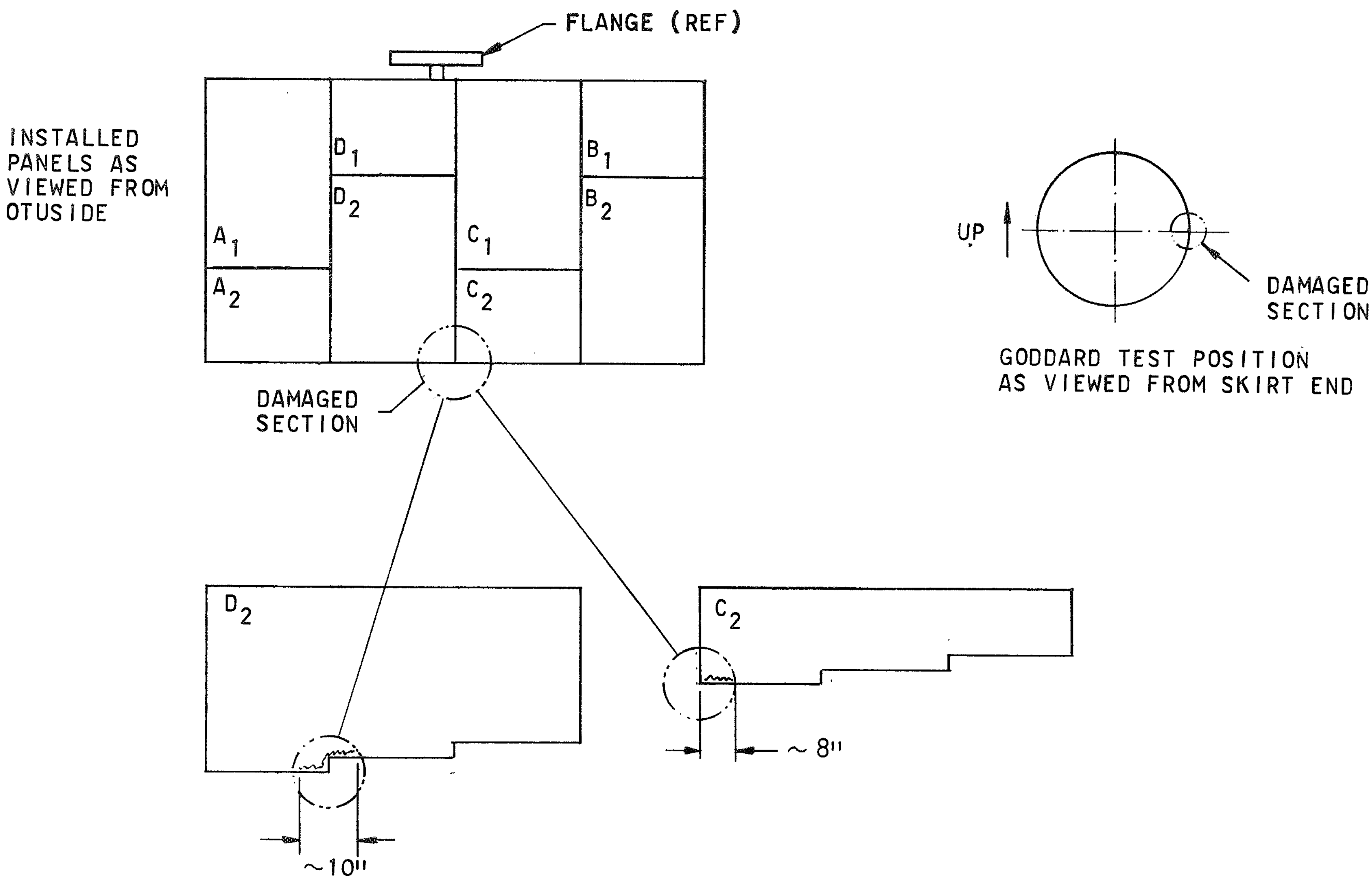


FIGURE 35 PANEL DAMAGE ZONE DEVELOPMENT
POST TEST EVALUATION

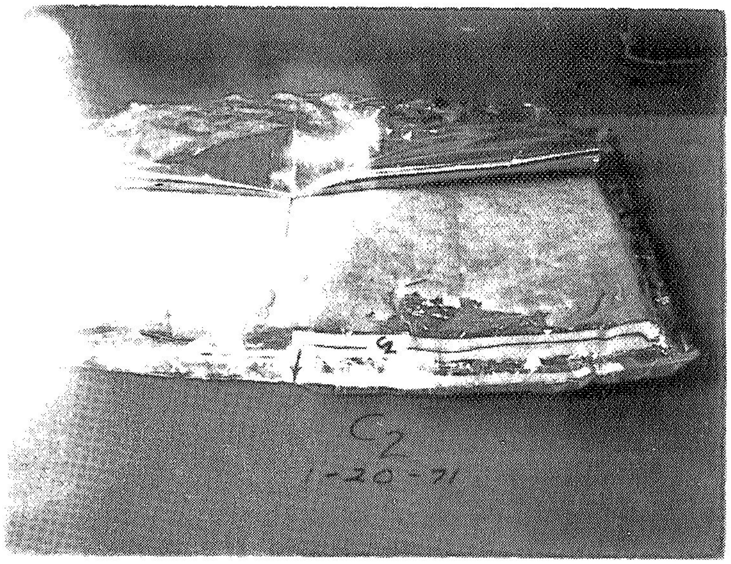


FIGURE 36: C_2 Panel Damage - As Viewed from the Tank Side

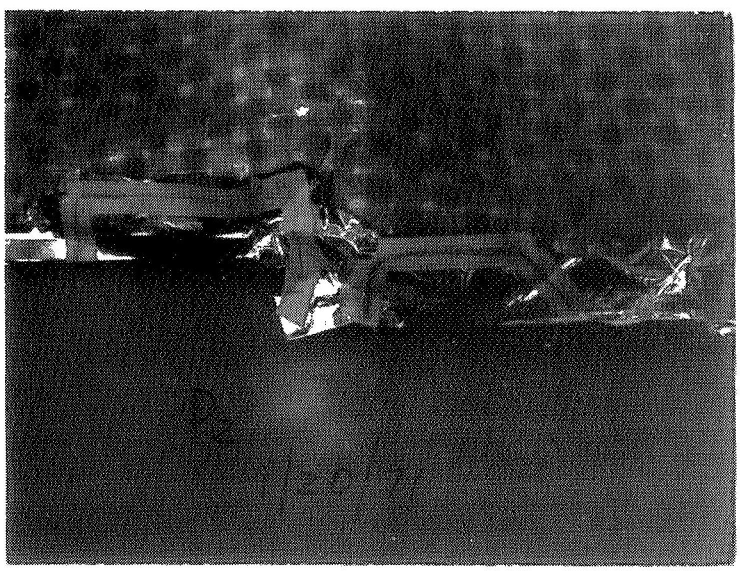


FIGURE 37. D_2 Panel Damage - As Viewed from Tank Side.

tank position. Location of the damaged panel joints are shown in Figure 35. The damaged position coincides with the area of debris observed in movies taken during the vibration testing at Goddard Space Flight Center. It seems certain that the loose foam came from panel C_2 as can be observed in Figure 36. Foam was observed to be missing from the three innermost layers. Panel D_2, although opened at the casing joint in a similar manner as C_2, suffered very little damage with only minimal breaking of the foam at wrinkles. (See Figure 37). The apparent reason for this was that the "D_2" panel was located at an inner layer, and was thus better protected (by the "C_2" panel). The "C_2" panel on the other hand was an outside panel, which therefore allowed this panel to move about more freely and probably was more thoroughly exposed during the accoustic noise tests.

The casing joint failure was a delamination of the 4 ply casing material. Thickness measurements of the casing indicated that one of the Mylar layers and the bond line remained with the other casing sheet, with delamination beginning at the vacuum draw radius, (formed to obtain the desired panel thickness). Actual joint thickness measurements for panel D_2 were .002 and .007 inch (5×10^{-5} and 1.78×10^{-4} M) respectively for the two sides. Basic material thickness is .0025 inch (6.3×10^{-5} M) while the measured joint thickness was .009 inch (2.29×10^{-4}M). Measurement of a similar area on the "C_1" panel indicated a joint thickness of .0075 to .009 inch (1.9×10^{-4} to 2.29×10^{-4}M). The C_2 panel also indicated delamination failure, but could not be measured, since this was an outside joint and therefore had been coated during installation and edge sealing.

Because the two panel failures occurred in close proximety to each other, the fact that both panels had joints delaminated, and the fact that the failed joint thickness was the same as that measured on another panel in a similar zone which did not fail, indicates that panel failures occurred because of installation or outside loadings and not because of faulty panel construction or materials. Possibly the failure is attributable to the fact that the backside of "C_2" and "D_2" panels had inadvertently been bonded to the fiberglas skirt with Narmco adhesive. This bond area extended beyond the casing joint, and therefore could have resulted in subjecting the joint to greater peel forces than anticipated if a pressure buildup occurred behind the panels. A VELCRO fastener was inadvertently located very close to the edge of the panel "D_2" and also may have contributed to the failure of the "C_2" and "D_2" panels. Panel "C_2" and "D_2" were attached to each other by this particular VELCRO. After the "C_2" panel failued, its movement and close attachment to "D_2" probably caused the "D_2" panel to fail.

The fact that a pressure buildup did occur was observed during testing and therefore lends strength to this theory. In fact, movies indicate that the pressure behind the panels was great enough to force the polar panels to bulge out in a convex shape, even though originally they were installed in a concave shape. Further proof that an over pressure existed is indicated by the fact that the three polar panels were literally ripped apart at the attachment points, with separation finally occurring at the closure bond lines rather than failing at the VELCRO closure. The forces involved resulted in tearing the foam, radiation shields and casing material as seen on Figure 38.

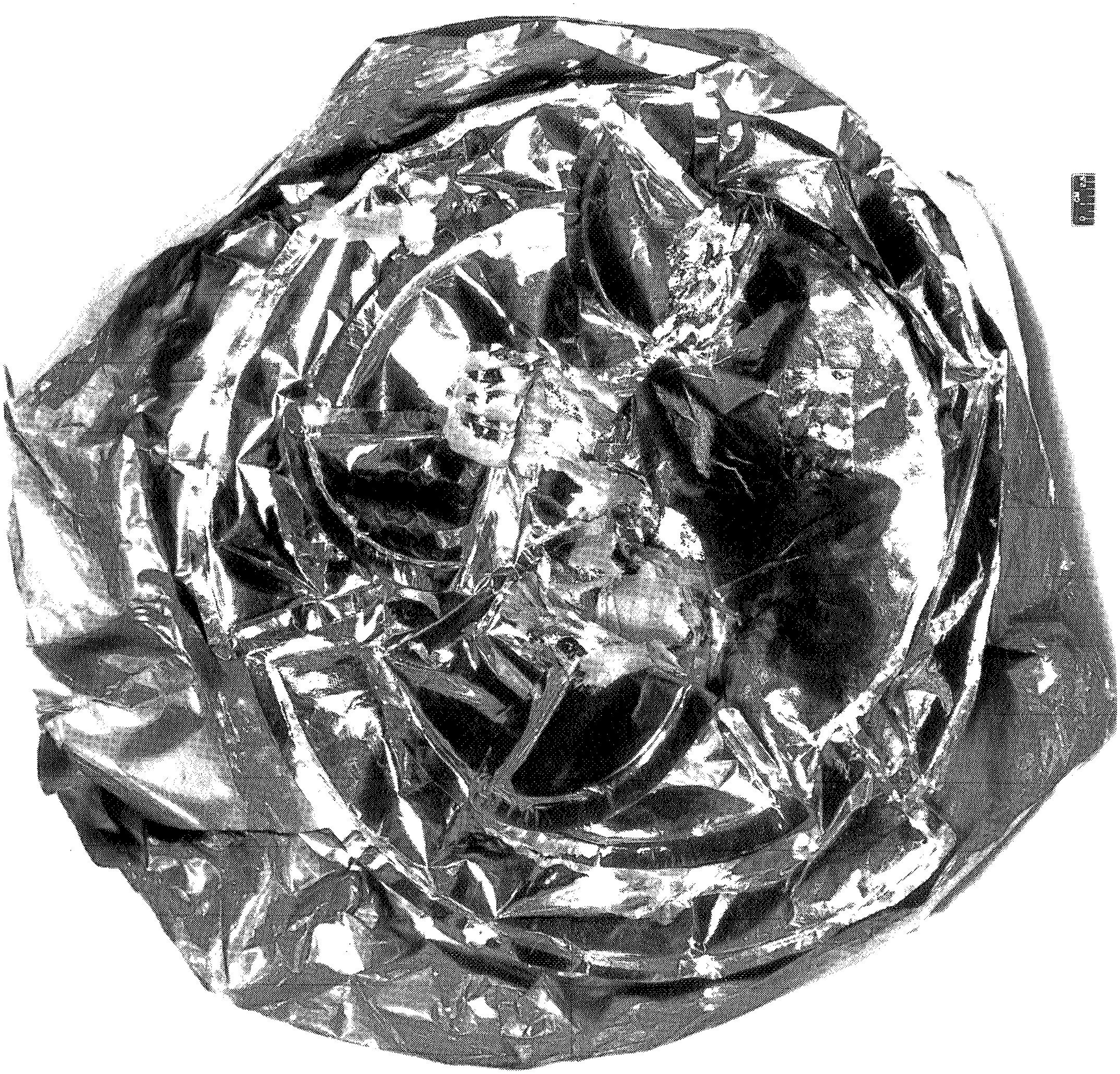


FIGURE 38. Polar Panels - As Viewed From Tank Side.
(C-71-340)

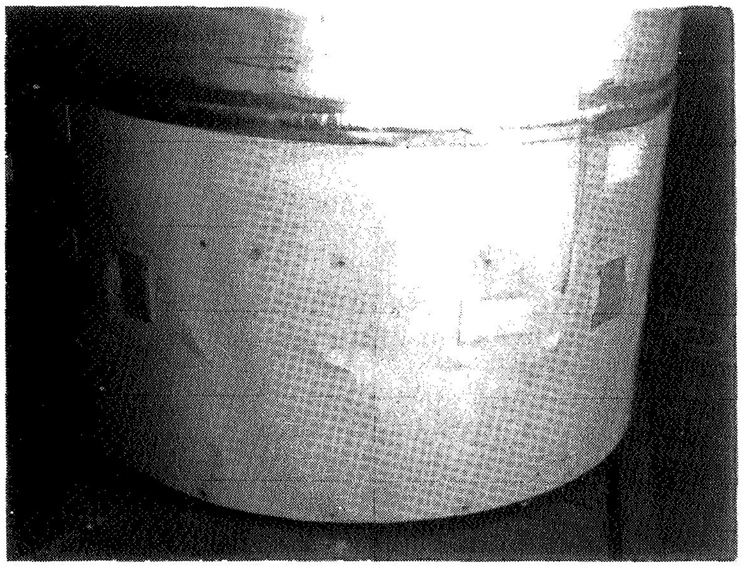


FIGURE 39. Damaged Fiberglas skirt Model Tank - Post Test Evaluation

The nine inner skirt panels were found to be in good shape and still attached to themselves and to the tank/inner skirt. The pillowing out of the polar panels as observed in movies of the tests did not involve the inner skirt panels. During post test examination of the test tank, it was also noted that the fiberglas skirt section had buckled. (See Figure 39.) The exact cause is not known, however it was likely caused by the vibration test specimen encountered during the offboard vibration test at Goddard, or possibly combined thermal and mechanical stresses. It is possible that these same test conditions caused the panels to fail, but not likely since the same panel configuration existed at three other locations in the SEMI system did not fail.

A helium leak test of the tank after the panels were removed, indicated that the test tank was still helium leak tight at ambient temperature. Machine sensitivity was established as 10^{-8} atm cc. helium per second.

The panel to panel seals were mechanically acceptable before and after physical testing. However, their performance in maintaining a leak tight space behind the panels is unacceptable in as much as on each cooldown cycle, seal damage was incurred which fortunately for this test did not require repair. Use of this sealing method would not be sufficient on a mission requiring more than one cycle such as a return to a 1 atmosphere external loading after the panels have been in the vacuum of space.

4.2.5 Materials

The list of all insulation system materials and the vendors used on the delivered insulation system is presented in tabular form, as Table 6.

TABLE 6

MATERIALS - LIGHT WEIGHT MODULAR MULTILAYER INSULATION
Contract NAS 3-12045

	Description/Use	Source
1. Panels		
Casing	Aluminized Mylar laminate - consisting of 4 - 1/2 mil Mylar Total thickness .0025 inch including adhesive	Dow Chemical Co., Buffalo, New York (Dobeckman Co.)
Foam Spacer	2 PCF Open cell rigid polyurethane foam. Special formulation	(1) Dacar Chemical Products Co., Pittsburgh, Pennsylvania (2) Union Carbide Chemicals South Charleston, West Virginia
	Foam Slicer	Tiffin Enterprises, Tiffin, Ohio
Getter	Palladium oxide	Union Carbide Corporation, Linde Division Tonawanda, New York
Radiation Shields	1/4 mil Mylar aluminized both sides to .5 $\Omega/\square$	Norton Co. - Metalized Products Division Winchester, Massachusetts
2. Miscellaneous		
Narmco 7343/7139	Cryogenic Adhesive	Crest Products Co., Santa Ana, California
Goodyear 207B 207C	Mylar Prime Coat	Goodyear Rubber Co., Akron, Ohio
Dow Corning Z 6020	Aluminum Prime Coat	Dow Corning, Cleveland, Ohio
VELCRO #65 Closure SA-0145A	Nylon Fastener	VELCRO Corporation, New York, New York
Carbon Dioxide - Coleman Grade	Cryopumping Gas	Matheson Chemical Co., E. Rutherford, New Jersey
Silicone Rubber	Panel-to-Panel Seals	General Electric RTV-102 New York, New York
'O' Rings	Viton-A O-ring/Seal off plate	Parker #2-24 Cleveland, Ohio

4.3 Subscale Testing

In order to achieve the objectives of the model system, small scale testing of the various concepts and subsystems were performed. The following sections describe the tests performed and the results achieved.

4.3.1 Small Panel Tests

Small panel testing consisted of vibration tests, evaluation of the effect of shrinkage due to evacuation and determination of the product (E I), (Youngs Modulus times the Moment of Inertia). Each is discussed in the following paragraphs.

Vibration tests were conducted on a 12 inch (.305M) x 24 inch (.61M) SEMI panel containing radiation shields and punched hole spacers with excellent results as there was no apparent damage. The tests were performed as per the subscale vibration test plan, included as Appendix 10 of this report. Photographs of the evacuated SEMI panel (Figure 40) and the recovered panel (Figure 41) are presented. A transparent casing material was used on the front of the panel to permit visual observation during tests. This same panel, shown in a vertical position in Figures 40 and 41 was also tested in two horizontal positions. The three positions are shown schematically in Figure 42.

The two dummy panels and the test panel were attached to each other and to the test fixture by nylon VELCRO fasteners located on the pattern shown in Figure 43. The 1 inch x 2 inch (.025 x .05 M) strips of VELCRO No. 65 nylon closure were placed at right angles to each other such that cross mateing the hook and pile would result in a one square inch contact area. Fastener closure pressure was achieved by evacuating the panel assembly, thereby achieving a 15 psi (1.03×10^5 newtons per M^2) uniform compressive loading, such as will be the case in actual panel assembly. As discussed in the test plan, the edges of the test panel, and the VELCRO strips were bonded to the test fixture with contact adhesive.

As outlined in the test plan, the panel assembly was initially subjected to 1g sweeps from 20 to 150 Hertz and then increased to 8.5g sweeps in the range of 20 to 150 Hertz for evacuated and/or recovered panels. In addition to the stated test plan loads, the recovered panel assembly was held at a constant frequency of 55 Hertz and 8.5g for approximately 10 minutes in position B as shown in Figure 42, without any apparent damage. Total vibration testing of the panel assembly for all positions was approximately 50 minutes of which 45 minutes of testing was at the 8.5g vibration level. The foam spacer material appeared to be intact after testing, with no indication of powdering or foam separator breakage. The Mylar radiation shields and the casing also did not appear to have experienced any damage. As a final test, the two dummy panels were removed leaving only the test panel (not shingled) adhered to the fixture only along the bottom edge, and four VELCRO fasteners located at each corner. The test panel was then vibrated in position B, Figure 42 in the recovered condition. The VELCRO test panel attachments, subjected to 8.5g sweep in 20 to 150 Hertz range, were slightly loosened, although no permanent damage occurred and the panels were easily rejoined after the test. This test was considered more stringent since the recovered panel was not constrained by adjacent panels as are the panels which will be installed on actual tankage.

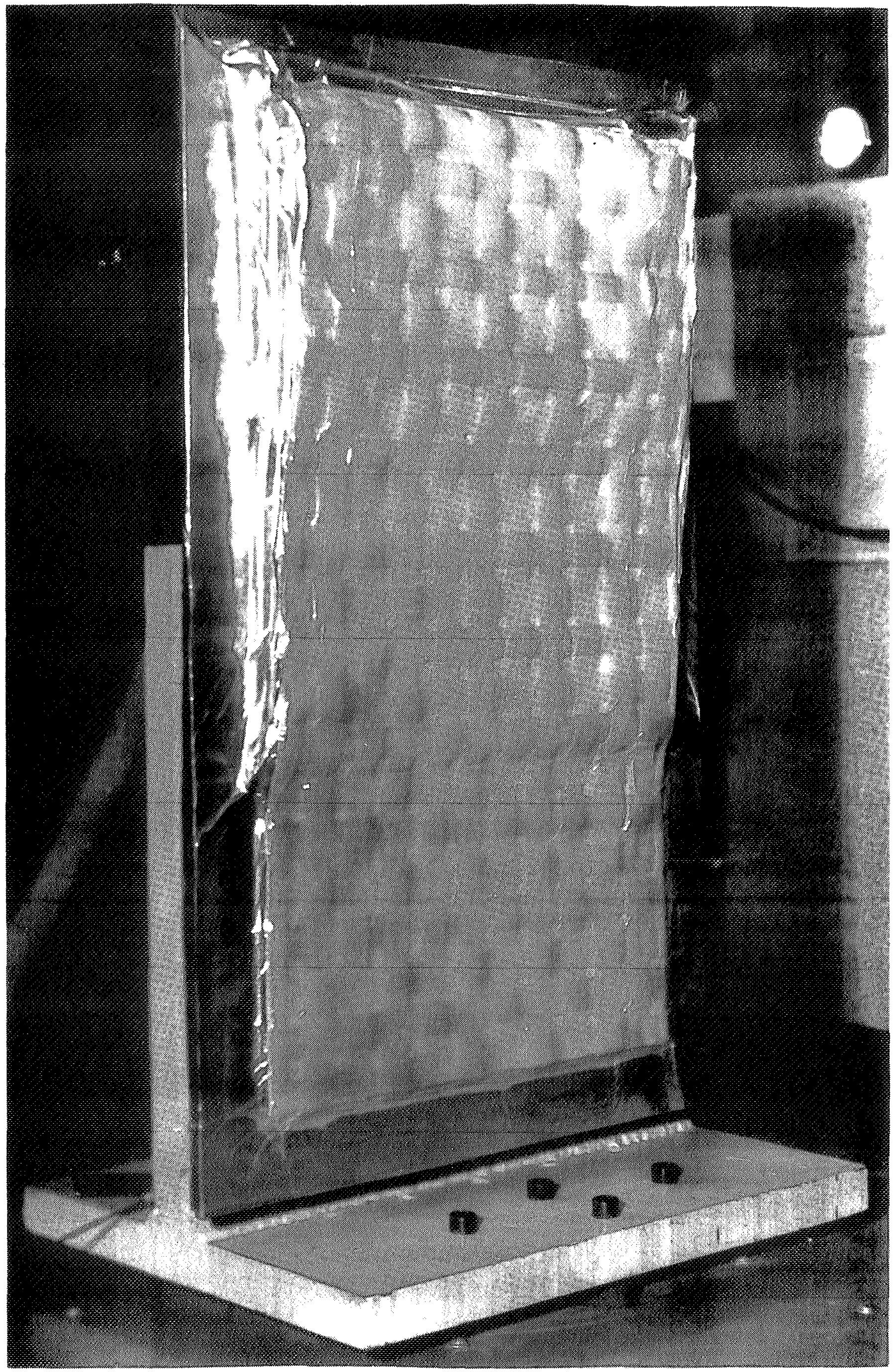


FIGURE 40. Evacuated Sub-Scale SEMI Panel Vibration Test (1125-69)

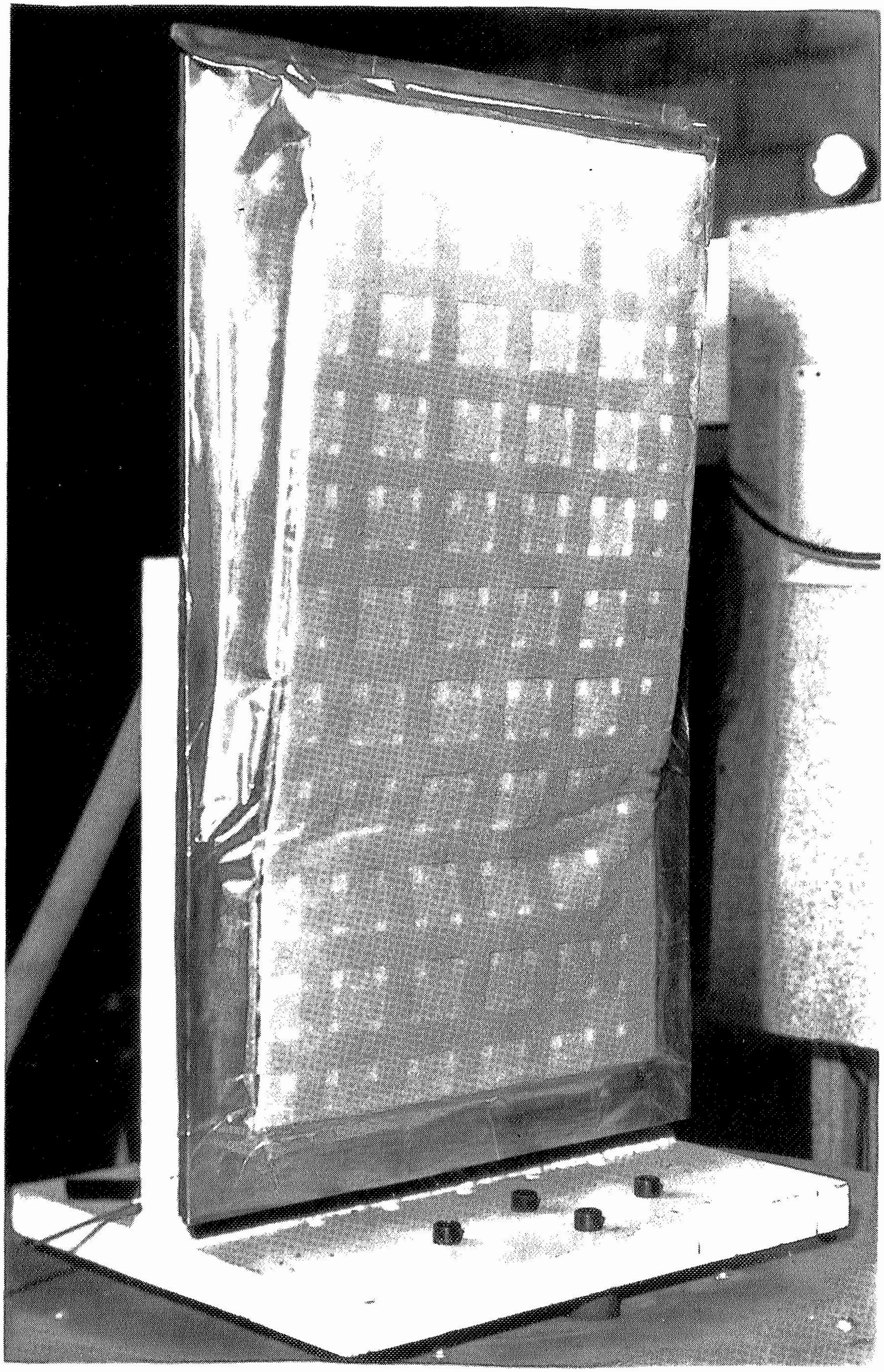
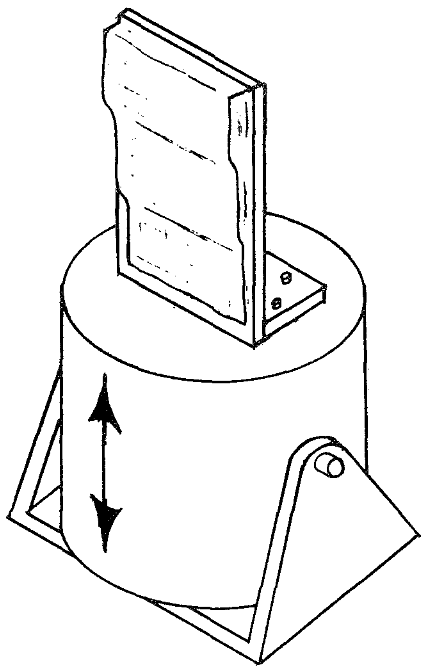
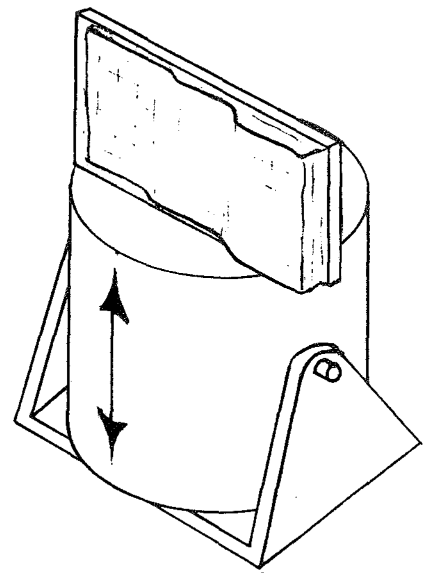


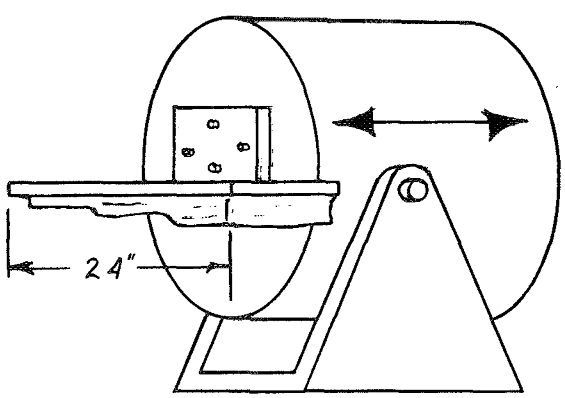
Figure 41. (Recovered) Sub-Scale SEMI Panel Vibration Test (1124-69)



A. Panel Vertical
Machine Head Vertical



B. Panel Horizontal
Machine Head Vertical



C. Panel Horizontal
Machine Head Horizontal

FIGURE 42. Subscale Panel Vibration Test Positions

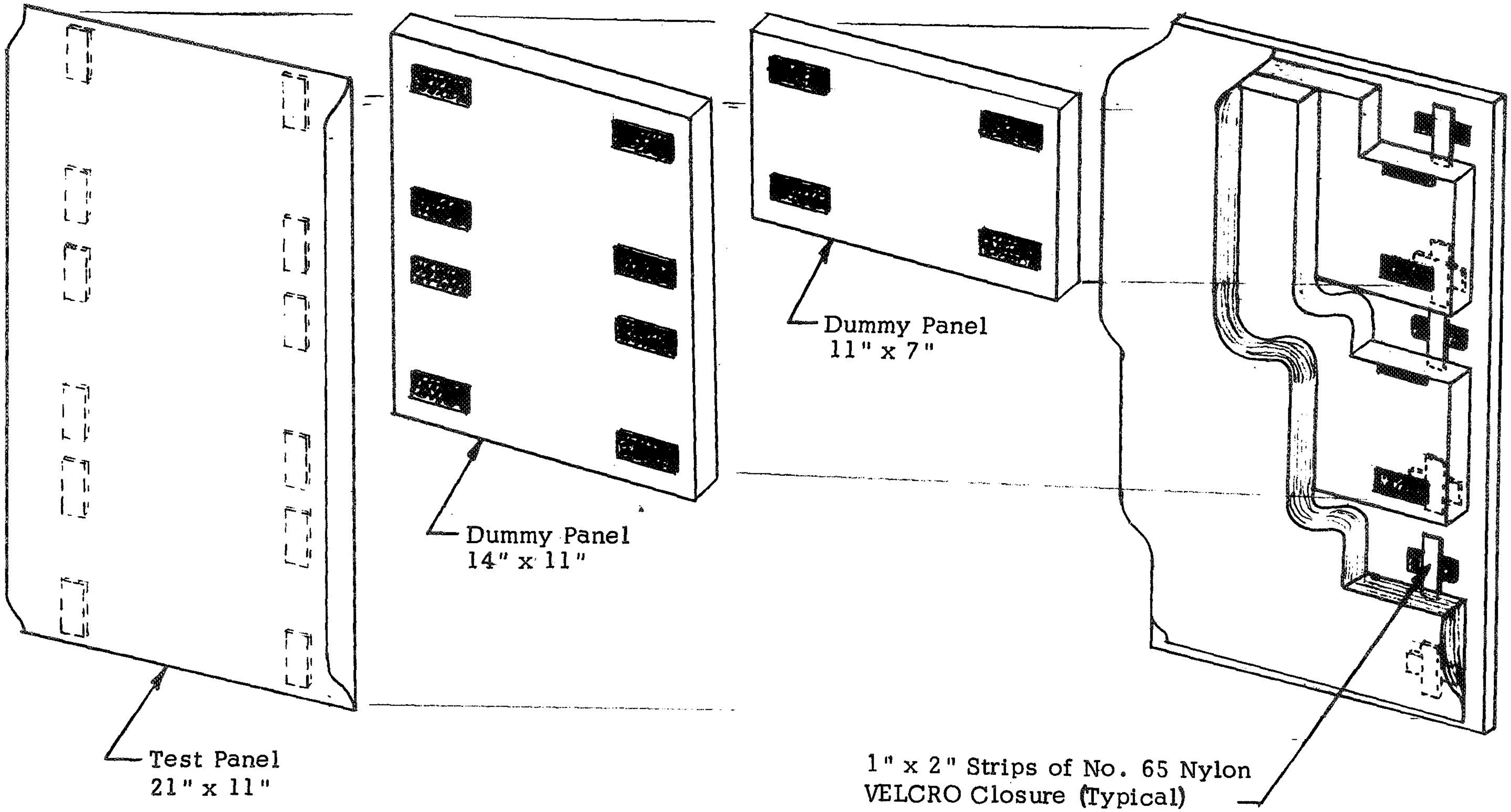


FIGURE 43. VELCRO Fastener Placement
Subscale Test Panel

Additional testing involved measuring the amount of shrinkage a panel undergoes during evacuation. This shrinkage was determined to be .12 inches (3×10^{-3}M) in 20 inches (.51M) or 0.006 in/in. (M/M). This was determined by measuring the distance between two points on the casing material before and after evacuation of the unrestrained panel. This shrinkage, attributable to evacuation, is not permanent. It is likely a result of the punched foam spacer being slightly corrugated due to irregular number of layers in the composite spacer. This corrugated surface is somewhat visible in the transparent casing shown in Figure 40.

The product EI (Young's Modules times the moment of inertia) for the unevacuated PT-6 SEMI panel configuration was determined to be 14.33 lb-in^2) 9.9×10^3 N/M^2 for the vibration test panel. This value was calculated from a free cantilever deflection of 3-1/4 inches (.0825M) measured 14 inches (.355M) from the support table, for a 1/2 inch (.0127M) thick foam panel, 11 inches (.279M) wide.

4.3.2 Foam Spacer Evaluation

Urethane foam for previous contract work on the SEMI system has been obtained from Union Carbide Chemicals as an experimental item. For this contract, it was decided to evaluate a commercial foam vendor, in addition to Union Carbide Chemicals. Foam was purchased from Dacar Chemicals Company and also Union Carbide Chemicals. The foam was produced by both Companies according to the following formula: (previously used in contract NAS 3-6289 and NAS 3-7953).

TABLE 7

UCC - Open Cell Rigid Polyurethane Foam Components

NIAX Polyol T-221	100.0 PBW
Ucon - 11	30.0
L-5320	4.0
TMBDA	0.6
Aluminum No. 422	1.0
NIAX AFPI	103.0
Stannous Octoate	0.2

Compression tests performed at ambient temperatures of foam cubes in the same manner as in the previous contracts indicated both foams to be acceptable. The Dacar Chemical's foam however, was more compressible, probably attributable to a somewhat larger cell size. The larger cell size did limit the minimum sliced foam thickness to 0.035 inch (8.9×10^{-4} M). UCC foam could be sliced to .02 inch (5.08×10^{-4} M) thick.

In order to evaluate a multi-layered panel using the two foams, a small test panel, 11 in x 17 in (.279 x .431M) was fabricated containing two 8-1/2 in x 11 in (.216 x .279M) stacks of composite layers of foam and radiation shields. One stack consisted of UCC open cell foam, .02 in. thick (5.08×10^{-4}M) as used on the previous contract, while the other stack contained foam spacers, .035 inch thick (8.9×10^{-4}M) fabricated from the Dacar foam. Evacuation tests of the panel, using compressed and recovered dimensions of both composite stacks as criteria, indicates little difference between the foams in terms of load deflection characteristics or panel thickness after the initial evacuation. Prior to the initial evacuation, the Decar foam panel was thicker than the UCC foam panel as would be expected.

To determine the foam density, the samples were measured with vernier calipers, and the weight of each sample was obtained on a beam balance. The density was then calculated by dividing the sample weight by the volume. Density for both foams was in the range of 2.0 PCF (32 kg/M^3).

4.3.3 VELCRO Fastener Evaluation

To gain further experience and knowledge with the VELCRO fastener, several small scale static tests were performed. These included the determination of the tension, shear, and peel strengths of the nylon closure, as well as the associated attachment problems in bonding the nylon closure to Mylar and aluminum. This data was used in the panel structural analysis of Task I and Task II. (Section 4.1.3)

Limited testing of No. 65 Nylon VELCRO fasteners at ambient temperature and after LN_2 temperature cycles, indicates that the vendor coated fastener (VELCRO style No. SA 0145A - Methyl Ethyl Ketone (MEK) activated) was suitable for bonding nylon VELCRO fastener to both Mylar and aluminum substrates. While it has been determined that stronger bonds were obtained using a Goodyear prime and Narmco urethane adhesive, it was also apparent that the VELCRO loop and pile closure separated at a peel strength of less than 1 lb. per inch (1.79×10^4 gm per M), of width of closure, and did not require the stronger bonds. Indeed during the Mylar laminate casing material tests, laminate failure was experienced before failure of the adhesive joint being tested.

VELCRO samples, bonded to substrates, were subjected to six ambient cryogenic temperature cyclings before peel testing. Minimum 90° peel strengths of 4 lbs. per inch (7.16×10^4 gm per M) were measured on the VELCRO adhesive backed samples attached to aluminum plates. Separation occurred between the nylon fastener and the adhesive, with ~50% of the adhesive remaining on the aluminum. Similar 90° peel tests of the uncoated Nylon VELCRO fastener adhered to aluminum plates with the Goodyear/Narmco system yielded test results of 22 lbs. per inch (3.94×10^5 gm per M). Separation occurred between the Goodyear prime and the Narmco urethane adhesive. 90° peel tests performed using the Mylar casing materials as substrate resulting in delaminating the casing material regardless of the adhesive system used.

Shear tests performed on samples of the VELCRO adhesive to aluminum, and the VELCRO Goodyear/Narmco adhesive to aluminum, yielded values of shear stress in excess of 12.5 psi (8.6×10^4 newtons per M^2) without failures.

Shear tests of the assembled VELCRO fasteners (VELCRO loop to VELCRO pile assembled under a 15 psi (1.03×10^5 newtons per M^2) compressive load prior to testing, resulted in a joint capable of achieving 4.25 psi ($2.92 \times 10^{+4}$ newtons per M^2) shear stress, with approximately 0.2 inch (5×10^{-3} M) slippage occurring before separation. The complete shear stress vs. deflection curve for a 2 in. x 2 in. (.05 x .05M) sample of No. 65 VELCRO closure is shown in Figure 44. Results of other tests are shown in Table 8.

Tension tests of VELCRO loop to VELCRO pile assembled under various closing pressures indicated that joint strength is independent of closing pressure in the range of 15 to 50 psi (1.03×10^5 to 3.4×10^5 newtons per M^2), and therefore one atmosphere is sufficient to achieve closure. Joints tested indicated tensile strength on the order of 1 psi (6.87×10^3 newtons per M^2). Although this value is quite low and much less than the vendor published data, it is sufficient for this application.

4.3.4 Radiation Shield Resistivity

The object of this test was to determine the resistivity of the aluminized Mylar shield material to be used on this contract. Resistivity is an indication of emissivity for a particular vendor. The procedure, which is quite simple, involves measuring the resistance of the coating on each side over a given length, and then subtracting the end effects by measuring the resistance of 1/2 the initial length, and dividing by a dimensionless number per the following formula:

$$\overline{R} = \frac{(2R_t - R_1 - R_2)}{L} W$$

where R_t = Resistance of Total Length in OHMS

R_1 = Resistance of left half of total length in OHMS

R_2 = Resistance of right half of total length in OHMS

L = Total length (Between extreme contacts) = inch

W = Width of strip, inch

$\overline{R}$ = Resistivity - OHMS per square

The test sample in this case measured .25 inch (6.3×10^{-3}M) wide by 15 inches (.381M) long. Resistance measurements were taken between a 12 inch (.305M) gage length. The sample was clamped to a Plexiglas table using copper strips at either end to assure a good electrical contact over the total sample width.

The tests were performed separately, on both sides of the aluminized 1/4 mil (6.25×10^{-6}M) Mylar radiation shield. The measured resistance of the material used was 0.4 ohms per square while the contract required 0.5 $\pm$ 0.2 ohms per square. The resistivity in ohms per square is a recognized measurement technique.

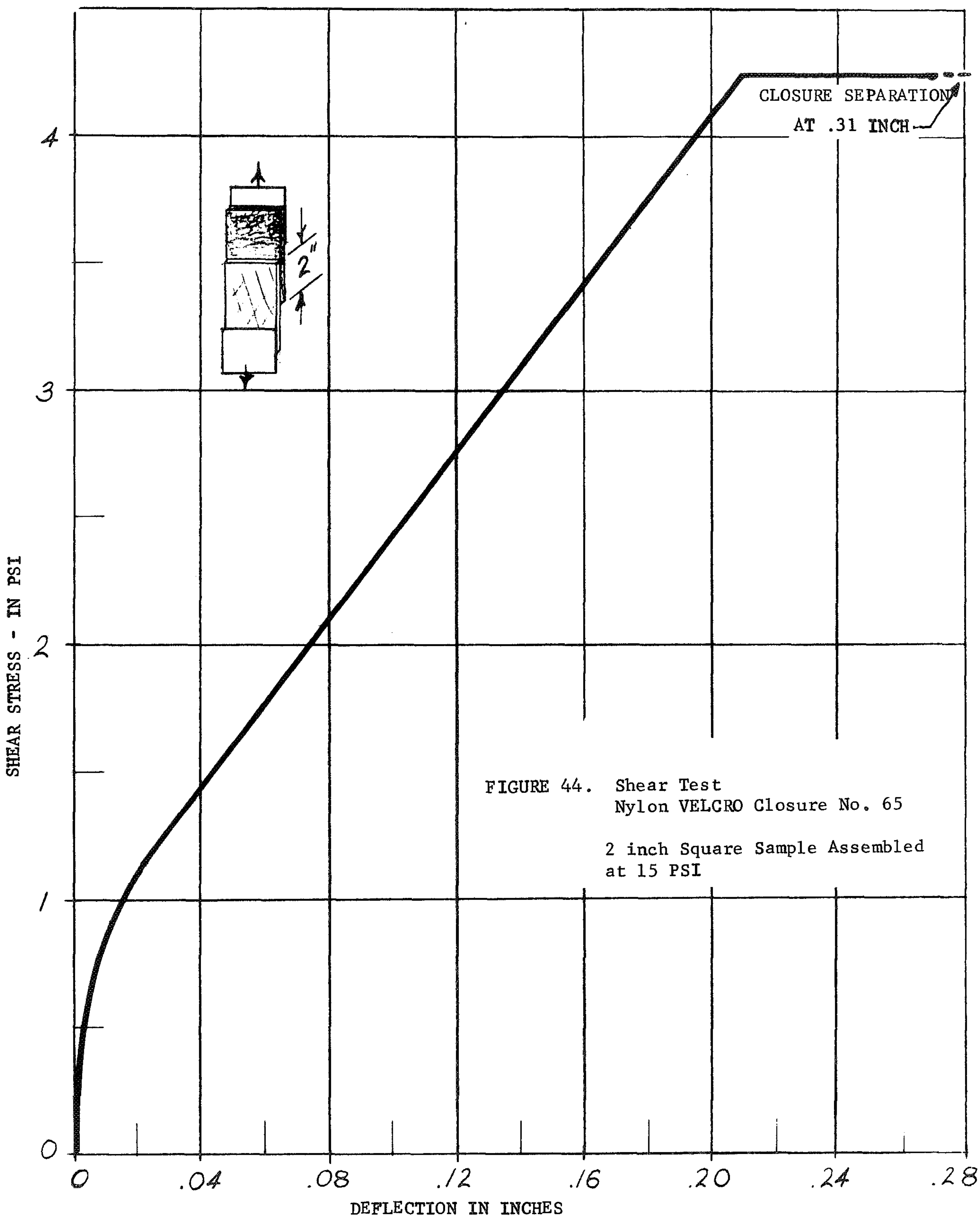
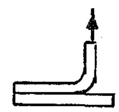
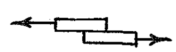
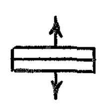


FIGURE 44. Shear Test
Nylon VELCRO Closure No. 65

2 inch Square Sample Assembled at 15 PSI

TABLE 8

SUMMARY - NYLON VELCRO TESTS

TEST SPECIMENS*	Quantity Tested	Results
Fastener 90° Peel Tests		
VELCRO to Aluminum		
VELCRO adhesive	4	~ 4 lbs. per inch to maintain 90° peel adhesive remained on aluminum.
VELCRO/Goodyear-Narmco	7	~ 22 lbs. per inch to maintain peel-separation occurred between Goodyear and Narmco
VELCRO to 4 ply laminate		
VELCRO adhesive	2	Casing material delaminate at ~ 4 lb. per inch
VELCRO/Goodyear-Narmco	3	Casing material delaminate at ~ 4 lb. per inch
Fastener Shear Tests		
VELCRO to aluminum VELCRO adhesive VELCRO/Goodyear-Narmco		Ultimate shear strength greater than 12.5 psi at ambient and cryogenic temperatures without failure
Closure Test- VELCRO to VELCRO		
Shear	 3	Separation at 4.25 psi at .31 deflection (See Figure 44)
Tension	 6	Failure at ~ 1 psi at ambient temperature

* Subjected to 6 temperature cycles from ambient to -320°F before testing.

4.3.5 Casing Material Permeability

Two casing material laminates were used on this contract. The interior portion of the panel, i.e. that not ordinarily exposed to atmospheric air consists of a 4 ply aluminized Mylar laminate, while the remaining portion of the panel is a composite of a Mylar-aluminum foil-Mylar laminated to the 4-ply material. The permeability rate of both materials is required. However for the laminate of foil and 4 ply laminate, since the permeability of the foil laminate is so much less than the permeability of the aluminized Mylar, only the permeability of the Mylar/foil laminate was evaluated to indicate the overall permeability of the foil/4 ply composite.

Permeability Test Air Permeation Barrier Film

Four, 6 inch (.15M) diameter samples of a Mylar-aluminum foil-Mylar casing material were helium leak chebked per procedures developed under previous contracts (See Appendix 12). The results of these four tests indicate that the material is acceptable for use as the barrier. (The material will be laminated to the four ply casing material at the air exposed outer 1/3 of the SEMI panel). It will be necessary to visually inspect the material for gross pinhole leaks. The results are listed below:

Test No.	1	Unable to pump below 50μ - pinholes
	2	Gross leak - No test
	3	Less than 2.2×10^{-8} atm. cc. helium per sec-ft^2 atm. (2.37×10^{-7} atm. cc. helium per sec. M^2 atm.)
	4	Less than 2.2×10^{-8} atm. cc. helium per sec-ft^2 atm. (2.37×10^{-7} atm. cc. helium per sec. M^2 atm.)

Permeability Test - 4 Ply Aluminized Mylar Laminate

Six inch (.15M) diameter samples of the 4 ply casing material were helium leak tested to determine permeability per procedures listed in Appendix 12. Because of the high permeability of this material and the sensitivity of the helium detector, it was necessary to use a special test gas mixture of 1% helium gas in nitrogen gas, as opposed to using 100% helium gas. By the use of this technique, the range of the leak detector can be extended, allowing the permeability to be determined by correlating the measured leak with the concentration of the helium in the test gas mixture, the leak rates being proportional to gas concentration.

The established leak rate for the material used on this effort was 3.5×10^{-6} atm. cc. helium per second-foot^2.

Page

5.0 REFERENCES

1. CR 72017 Investigation of a Lightweight Self Evacuating Prefabricated Multilayer Insulation System for Cryogenic Space Propulsion Stages by L. R. Niendorf and G. E. Nies (NAS 3-6289)

2. CR 72363 Lightweight Multilayer Insulation System by C. R. Lindquist and G. E. Nies (NAS 3-7953)

3. ASME Unfired Pressure Vessel Code Section VIII Division I

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5. Formulas for Stress and Strain Raymond J. Roark, McGraw Hill Book Co., Copyright, 1965.

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APPENDIX 1

SEMI Panel Stress Analysis

Full Scale Insulation System

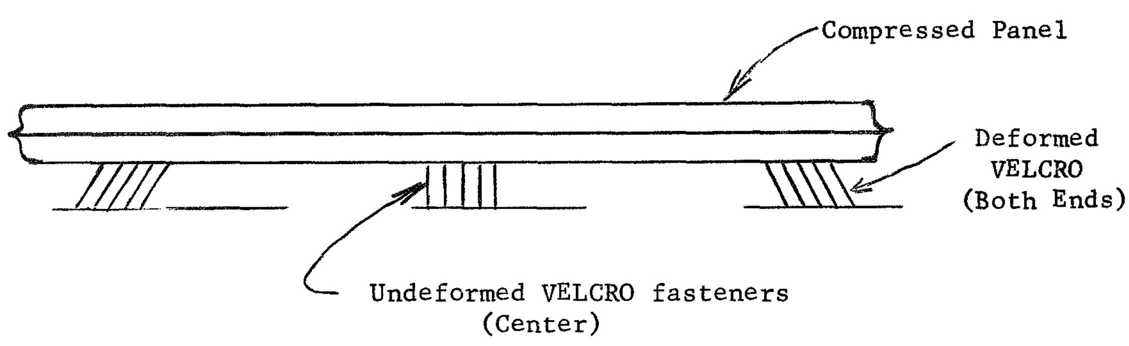
1. Thermal Stresses

Tank material: aluminum
Panel skin material: aluminized mylar

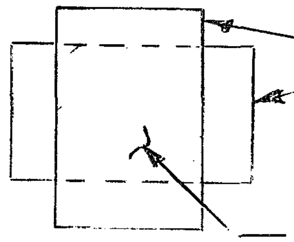
The contraction of aluminum shell from 293 to 20°K is 4.15×10^{-3} in/in. Average coefficient of thermal expansion for mylar film between 293 and 77°K is 2.22×10^{-5} in/in. °K. The value of additional contraction from 77 to 20°K is not available for mylar, but it is estimated to approach zero, particularly when compared to evacuation contractions which were demonstrated to be .00625 in/in. of panel. Therefore, it is not necessary to perform the dynamic tests at Goddard using liquid helium. Thus, assuming that 4.80×10^{-3} in/in. represents the total contraction of Mylar from 293°K to 20°K, there is a differential contraction equal to 0.65×10^{-3} in/in. which is active only on the portion of the panel attached directly to the tank. If VELCRO fasteners are used they are expected to provide the necessary motion without large resulting forces. On the other hand, if adhesive is used, stresses may develop in the panel skin, since the adhesive is expected to be quite rigid at cryogenic temperatures.

a. Stresses Resulting from Thermal and Evacuation Contractions

Assuming that an 8' by 8' panel contracts towards the center the total contraction near the extremities of the panel will be 48(.00065 + .00625 = .331 in.



Some side slip in the VELCRO closure starts, to occur at a deformation of .21 inch. This means that some slipping will occur at the outer attachment points. If excess area is provided such slipping will occur under constant force and once the contraction is accomodated the attachment will remain under their maximum stress which, from test is 4.25 psi.



At the attachment points Mylar skin will be in tension. If we assume the attachment areas to be 2 in. square, the force per joint

$$F = 4.25 \times 4 = 17.0 \text{ lbs.}$$

and the stress in Mylar is

$$S_{mylar} \ \frac{17.0}{3 \times .002} = 4250 \text{ psi}$$

This is a safe value for Mylar. As long as there is an external force compressing the panels, the same pressure will act on the whole structure, since the space between the panels will also be evacuated. The coefficient of friction for Mylar is found to be about 0.4 so that the panels are held from slipping with a frictional force of 14.7 x .4 = 5.88 psi. For an 8 ft. panel to slip would require a tensile stress in Mylar equal to

$$S_{mylar} = \frac{48 \times 5.88}{.002} = 141{,}000 \text{ psi}$$

which would cause a failure in Mylar. On the other hand, if the Mylar is held in place without slipping, the resulting strain due to thermal contraction and evacuation is .0069 in/in. This, from test, corresponds to about 8,000 psi stress, well below the breaking strength in Mylar.

Since now the possibility is admitted of the panels being held fast by the frictional forces it is necessary to check whether foam can accept strains due to thermal contraction without breaking. The average coefficient of thermal expansion for polyurethane foam in the region of 293 to 77°K is 6.39×10^{-5} in/in. °K which means that the total contraction is 1.380×10^{-2} in/in. Since the strain at failure for polyurethane foam is about 4×10^{-2} in/in. the foam should take the strain due to thermal contraction without cracking.

However, regardless of the strength of the materials involved, both of these failures appear to be physical impossibilities if the time element and change in temperatures are evaluated on a real time.

b. Dynamic Loads

The maximum axial dynamic load is 8.5 g. This load will place the side insulation in shear and the head insulation in tension or compression. The panels weigh 2.5 lb/ft^3 recovered. Since the thickness of recovered panels is 0.5 in. and we consider three thicknesses of panel to be attached to the tank, the total stress is

$$W = \frac{2.5}{144} \times \frac{.5 \times 3 \times 8.5}{12} = 0.0182 \text{ psi}$$

W is the dynamic force per unit area exerted as shear on the side insulation and in tension or compression on the head insulation. Since, due to atmospheric pressure, there will be a force of 14.7 psi normal to surface and 5.88 psi tangential (friction force), there will be no slippage or sagging of the insulation panels.

The maximum transverse dynamic force is 1.5 g. The same arguments as above hold, so that no difficulties should be anticipated.

2. Stress Behavior of Panels While they are Cold but Recovered

With panels recovered, but with launching loads still imposed, the following modes of failure should be considered.

a. Differential thermal contraction

b. Shear load on side panels skin due to longitudinal loads (8.5 g max.)

c. Tensile load on head panels due to longitudinal loads.

d. Combined shear and tensile load on the side panels during axial acceleration and vibration.

e. Tendency to break the foam sheets in buckling due to longitudinal loads.

f. Stresses on panel casing material seals

a. Differential Thermal Contraction

Since there is only the differential thermal contraction acting on the panel the effective contraction at outer attachment points is .0312 in. (see 1a). From test, this is equal to 1.3 psi on the VELCRO joints, which is well within the capability of the closure.

b. Shear Load in Side Panel Skin Due to Longitudinal Load (8.5 g max.)

The force due to dynamic load is 0.0182 psi (see 1 b). Allowing 4 in^2 of VELCRO attachment area per 1 ft^2 of panel the shear stress in the VELCRO joint is $.0182 \times \frac{144}{4} = .66$ psi. Added to the thermal contraction stress of 1.3 psi, we have 1.96 psi total shear stress which is safe for VELCRO joints.

c. Tensile Load on the Head Panels due to the Longitudinal Loads

Since the panels are on the front end of the tank the steady acceleration load of 2g will cause the compression of the panels, however these forces will cause no problem because of the low panel density. The vibration loads will, of course, introduce both tensile and compressive stresses, but the response of the recovered panels to vibrations in the 20-150 hertz range appears to be so low that no problems are anticipated with this type of loading.

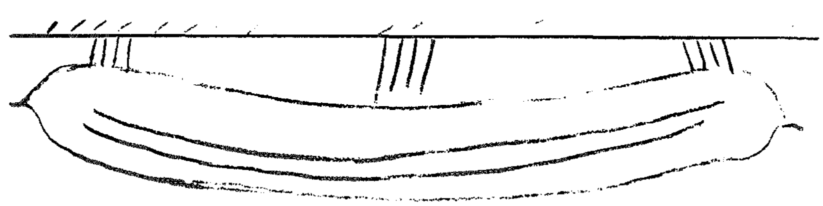
d. Combined Shear and Tensile Load on the Side Panels During Axial Acceleration and Vibration

This load can occur in the following combinations:

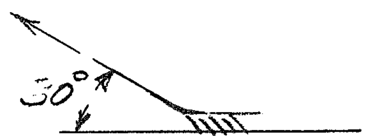
Shear load: 8.5 g, tensile load: 1 g
Shear load: 2.5 g, tensile load: 1.5 g

For 8.5 g shear load it was found in 2b that the total shear stress is 1.96 psi. In itself this stress is satisfactory for VELCRO fasteners.

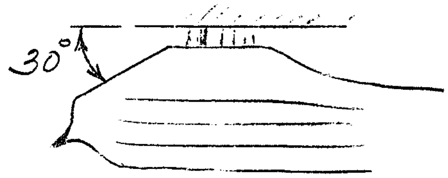
The tensile load will not be distributed uniformly over all the attachments.



As shown in the figure, the contents of the panel will exert a pressure on the outer skin which will then be transmitted through panel edges to the inner skin. From this it follows that instead of being evenly distributed over all the attachment points the load will exert a pull on the outer edges of outer attachments creating a peeling effect. The peeling of the joint, however, cannot occur unless the force acts



at an angle of at least 30° from the plane of the attachment. In order for the force to act at that angle a certain sagging of the panels must occur.



It may be noted that the cylindrical panel system has a fixed outer circumference resulting from panels being sealed at their outer edges. The upper half of the panels compresses somewhat under its own weight and the weight of the lower panels hanging on them. Under 1.5 g load this amounts to

$$0.0182 \times \frac{2 \times 1.5}{8.5} = .0065 \text{ psi}$$

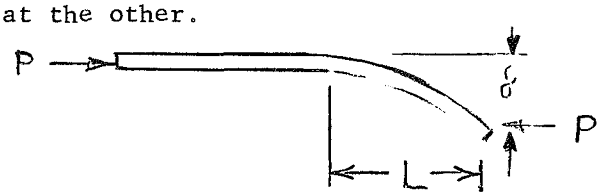
and the compression corresponding to this pressure can be neglected. At the bottom the panels will compress under their own weight x 1.5 g which is also negligible. If the attachment is, say, 1 inch from the edge of the panel, in order to achieve a 30° angle with the attachment the deflection δ is

$$\delta = \ell \sin 30° = 1 \sin 30° = 0.5 \text{ in.}$$

Since such deflection cannot be attained, the panel contents will rest on the outer skin before any peeling on the VELCRO fastener occurs. It is, however, recommended that VELCRO attachment be placed at least 2" away from the edge of panel as a precaution.

e. <u>Tendency for the Foam Sheets to Break in Buckling Due to Longitudinal Loads</u>

The foam sheets inside the panel may buckle when exposed to longitudinal loads. The most severe case will be buckling of the end of the panel which will be equivalent to a column built-in at one end and free at the other.



The buckling condition for such a configuration is

$$P = \frac{\pi^2 EI}{4L^2}$$

or

$$L = \frac{\pi}{2}\sqrt{\frac{EI}{P}}$$

From test EI = 1.30 in^4-psi/in. of panel width. Assume P to be the weight of a strip 20 ft. long x 1 in. wide x .5 in. thick multiplied by 8.5 g.

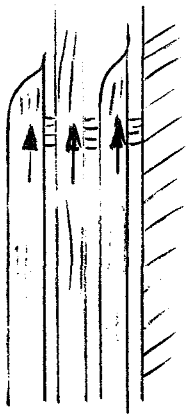
$$P = \frac{2.5 \times 8.5 \times .5 \times 20}{144} = 1.48 \text{ lb.}$$

$$L = \frac{\pi}{2}\sqrt{\frac{1.30}{1.48}} = 1.47 \text{ in.}$$

Thus, the smallest length to buckle is 1.47 in. To cause breaking of the foam substantial deflection δ (higher than 0.5 in.) will be needed for this length. Again, because of the fixed outer circumference the deflections will be limited to small values, so that buckling is not expected to be a problem.

f. Stress on Panel Casing Seals

The most serious stress appears to be the peeling stress on the panel seal which could come into being with the configuration of the panel as shown below.



For an 8 ft. long panel the weight per inch of width is

$$\frac{3 \times 2.5 \times 8 \times .5}{144} = 0.208 \text{ lb./in.}$$

With 8.5 g $P = 1.77$ lb/in. of seal. Since the seals can withstand greater forces, this is a satisfactory condition.

APPENDIX 2

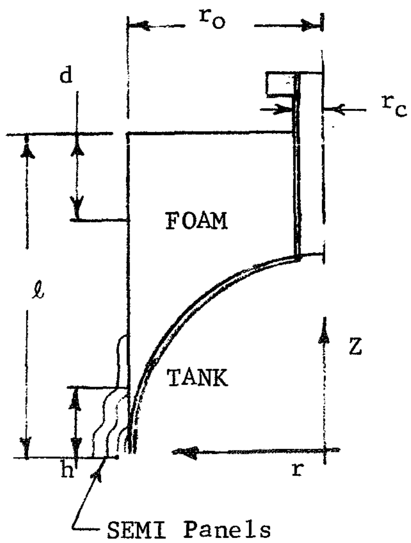
SUPPORT END INSULATION THERMAL ANALYSIS

The upper end of the Model tank is not insulated with a staggered polar panel system, as is the lower end. Instead, it is surrounded by a cylindrical plug of foam. This foam is isotropic and of uniform thermal conductivity k.

To evaluate the temperatures and heat fluxes within this foam plug, the temperatures at all surfaces were prescribed as follows:

$$(T = T\,(r,z))$$

Top Surface:	$T\,(r,\ell) = T_w = T_{wall}$
Inner Pipe:	$T\,(r,z) = T_c = T_{cryogenic}$
Spherical Insert:	$T(r, \sqrt{r_o}, z) = T_c,\; Z \leq h$
	$= T_c + (T_w - T_c)(\frac{z}{\ell - d}),\; h < Z \leq (\ell - d)$
	$= T_w \quad Z > (\ell - d)$



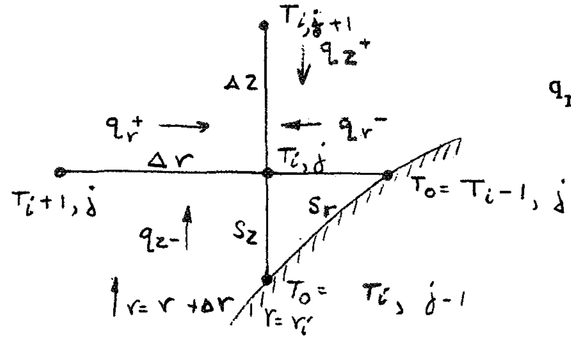
One can see that, at the outer surface, the temperature is prescribed at cryogenic for a certain height h, then increases linearly to ambient, remaining at ambient for a distance and at the other end of the plug.

The Fourier equation in cylindrical coordinates is

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial Z^2} = 0 \qquad (1)$$

Partly because of the geometric complication introduced by the hemispherical insert at the bottom of the plug, a simple relaxation technique was chosen for solving the equations.

Since the system is at steady state, the sum of all heat fluxes entering any point must vanish. Thus, from diagram at the left:



$$q_{r+} + q_{r-} + q_{z+} + q_{z-} = 0 \qquad (2)$$

where $q_{r+} = a_{r+}\,(T_{i+1,j} - T_{i,j})$

$q_{r-} = a_{r-}\,(T_{i-1,j} - T_{i,j})$

$q_{z+} = a_{z+}\,(T_{i,j+1} - T_{i,j})$

$q_{z-} = a_{z-}\,(T_{i,j-1} - T_{k,j})$

and

$$a_{r+} = \frac{2\pi k \Delta z}{\ell n(1 + \frac{\Delta r}{r_i})}$$

$$a_{r-} = \frac{\pi k\ (\Delta z + Sz)}{\ell n\ (\frac{r_i}{r_i - s_r})}$$

$$a_{z+} = \frac{2\pi k\ r_i\ \Delta r}{\Delta z}$$

$$a_{z-} = \frac{\pi k\ r_i\ (\Delta\ r + S_r)}{S_z}$$

By setting up a finite-difference mesh within the plug, setting up a heat balance equation in the form of equation (2) at each point, and assuming an initial temperature at each point, we can sweep through all the mesh points repeatedly, using equation (2) to find $T_{i,j}$, until convergence is achieved. We then evaluated the heat fluxes from the converged temperatures as summarized in the general text.

DESIGN REQUIREMENTS:

GEOMETRY:

a. OVERALL DIAMETER 28 TO 36 IN.

b. OVERALL LENGTH 72 IN.

c. SUPPORT

TANK WILL BE SUPPORTED FROM ONE HEAD VIA A SINGLE ALUMINUM PIPE CONNECTED TO A STAINLESS STEEL TRANSISTION JOINT AND BOLTED FLANGE.

D. SUPPORT FLANGE

FLANGE WILL MATE WITH LIQUID HYDROGEN CALORIMETER SUPPLY AND SUPPORT STRUCTURE (NASA PLUMBROOK SKETCH * SK 681100) AND LINDE FABRICATED ADAPTER.

GENERAL SPECIFICATIONS:

a. TANK OPERATING PRESSURE RANGE

75 PSID INTERNAL TO 15 PSID EXTERNAL

b. TEMPERATURE RANGE

AMBIENT TO −423°F

c. MATERIAL 5083-O ALUMINUM

d. HEMISPHERICAL HEADS

e. STRAIGHT SKIRT SECTION AT END OPPOSITE SUPPORTING HEAD

f. TRANSITION JOINT

"TUBE-TURN" FT TYPE BETWEEN ALUMINUM TANK AND STAINLESS STEEL NECK TUBE WITH FLANGE

DESIGN CALCULATIONS CONTRACT No. NAS 3-12045 TANK DESIGN FOR EVALUATION OF SEMI PANELS	COMPUTED BY LKE DATE 9/69 REV. 2/71	REFERENCE DWGS.: SK-106298 SK-106297
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N.Y. 14152	CHK'D [illegible] APP'V'D [illegible]	SHEET NO. 1 OF 46 APPENDIX 3

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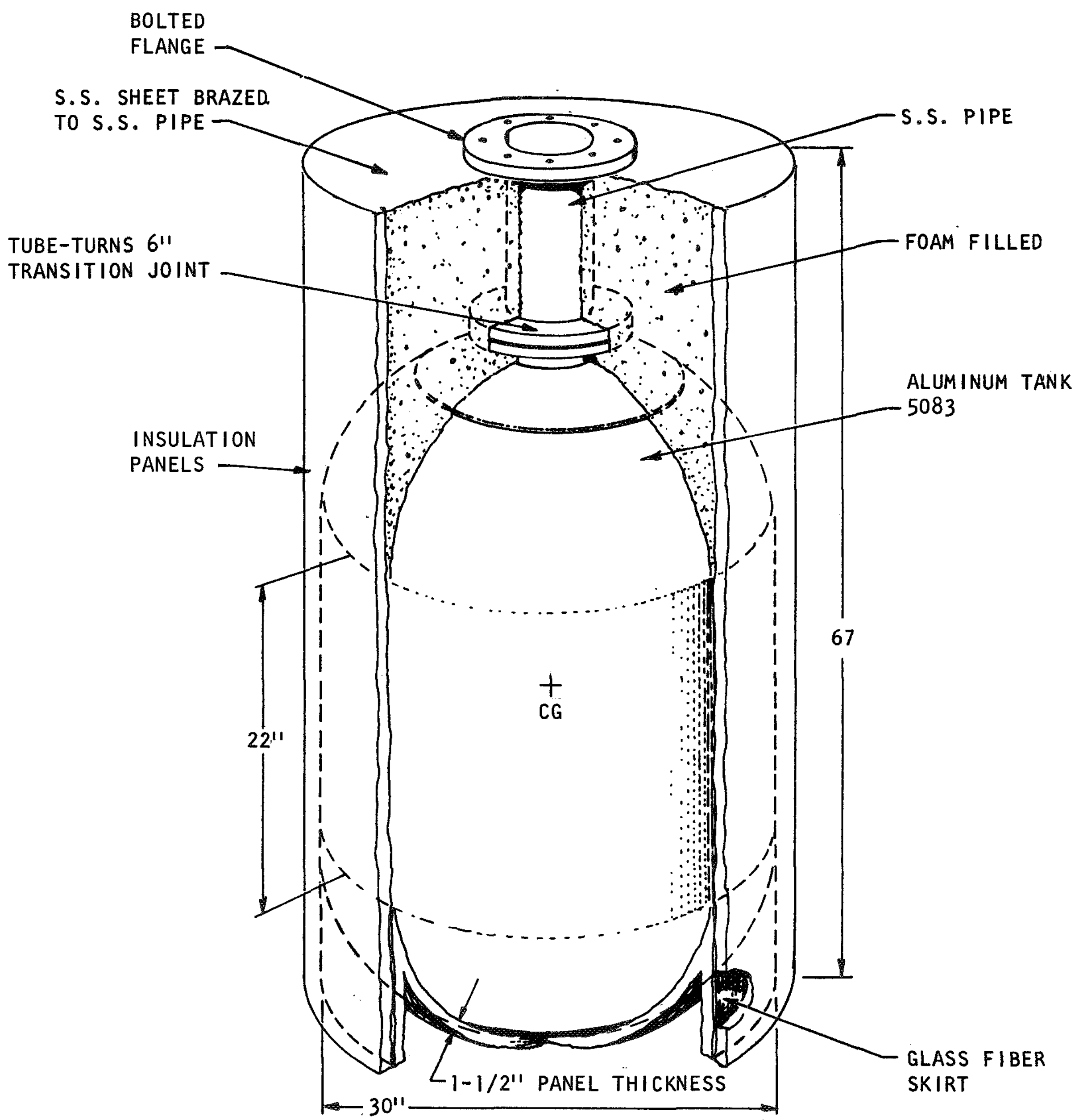
G. ALL WELDED CONSTRUCTION PER ASME BOILER AND PRESSURE VESSEL CODE - SECTION VIII, DIVISION 1.



DESIGN CALCULATIONS	COMPUTED BY LKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 2 OF 46
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VOLUME CHECK:

AN INITIAL CHECK OF THE TANK INTERNAL VOLUME IS MADE TO DETERMINE JURISDICTION OF THE CODES. (U-1, d-5) VESSELS MUST HAVE A NOMINAL WATER CONTAINING CAPACITY OF 120 GAL. OR MORE.

TWO 30" HEMISPHERES:

$$V = \frac{4}{3}\pi R^3 = \frac{4(3.14)(15)^3}{3} = 14130 \text{ in}^3$$

ONE 26" CYLINDRICAL SECTION

$$V = \frac{\pi D^2 L}{4} = \frac{3.14(30)^2(26)}{4} = 18369 \text{ in}^3$$

TOTAL VOLUME $= 32,499 \text{ in}^2 = 140$ gal.

∴ CAPACITY IS WITHIN JURISDICTION OF DIVISION 1, SECTION VIII

SHELL WALL THICKNESS:

DESIGN DATA

a.	EXTERNAL PRESSURE	15 psi
b.	INTERNAL PRESSURE	75 psi
c.	TEMPERATURE RANGE	AMBIENT TO −423°F
d.	MATERIAL	5083-O ALUM.
e.	MAX. ALLOWABLE STRESS (TABLE UNF-23, SPEC.# SB-209)	10,000 psi
f.	WELD EFFICIENCY (TABLE UW-12, NOT RADIOGRAPHED)	0.70

FOR INTERNAL PRESSURE (UG-27, Pg. 11)

CIRCUMFERENTIAL STRESS (LONGITUDNAL JOINTS)

$$t = \frac{PR}{SE - 0.6P}$$

WHERE
t = MINIMUM REQ'D THICKNESS
P = MAXIMUM DESIGN PRESSURE
= 75 PSI
R = INSIDE SHELL RADIUS
= 15 IN.
S = MAXIMUM ALLOWABLE STRESS
= 10,000 PSI
E = JOINT EFFICIENCY
= 0.70

SUBSTITUTING

$$t = \frac{75(15)}{10{,}000(.7) - .6(75)}$$

$$= 0.162 \text{ IN.}$$

NOTE:
@ P = 100 PSI., t = 0.250 IN., $\rightarrow$ S = 8660 PSI
$\therefore$ A 100 PSI BURSTING DISK WILL BE EMPLOYED

FOR EXTERNAL PRESSURE (UG-28, Pg. 11)

REFER TO FIG. UNF 28.23 (Sheet 5) FOR DETERMINATION OF SHELL THICKNESS UNDER EXTERNAL LOADING

WHERE
t = MINIMUM REQ'D SHELL THICKNESS
L = DESIGN LENGTH OF TANK
= DISTANCE BETWEEN HEAD-BEND LINES PLUS ONE-THIRD OF EACH HEAD DEPTH
$= 24'' + \frac{1}{3}(15) + \frac{1}{3}(15) = 34$ IN.
D_o = OUTSIDE SHELL DIAMETER
= 30 IN.
P = EXTERNAL DESIGN PRESSURE
= 15 PSI.

DESIGN CALCULATIONS	COMPUTED BY: XXE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 4 OF 46
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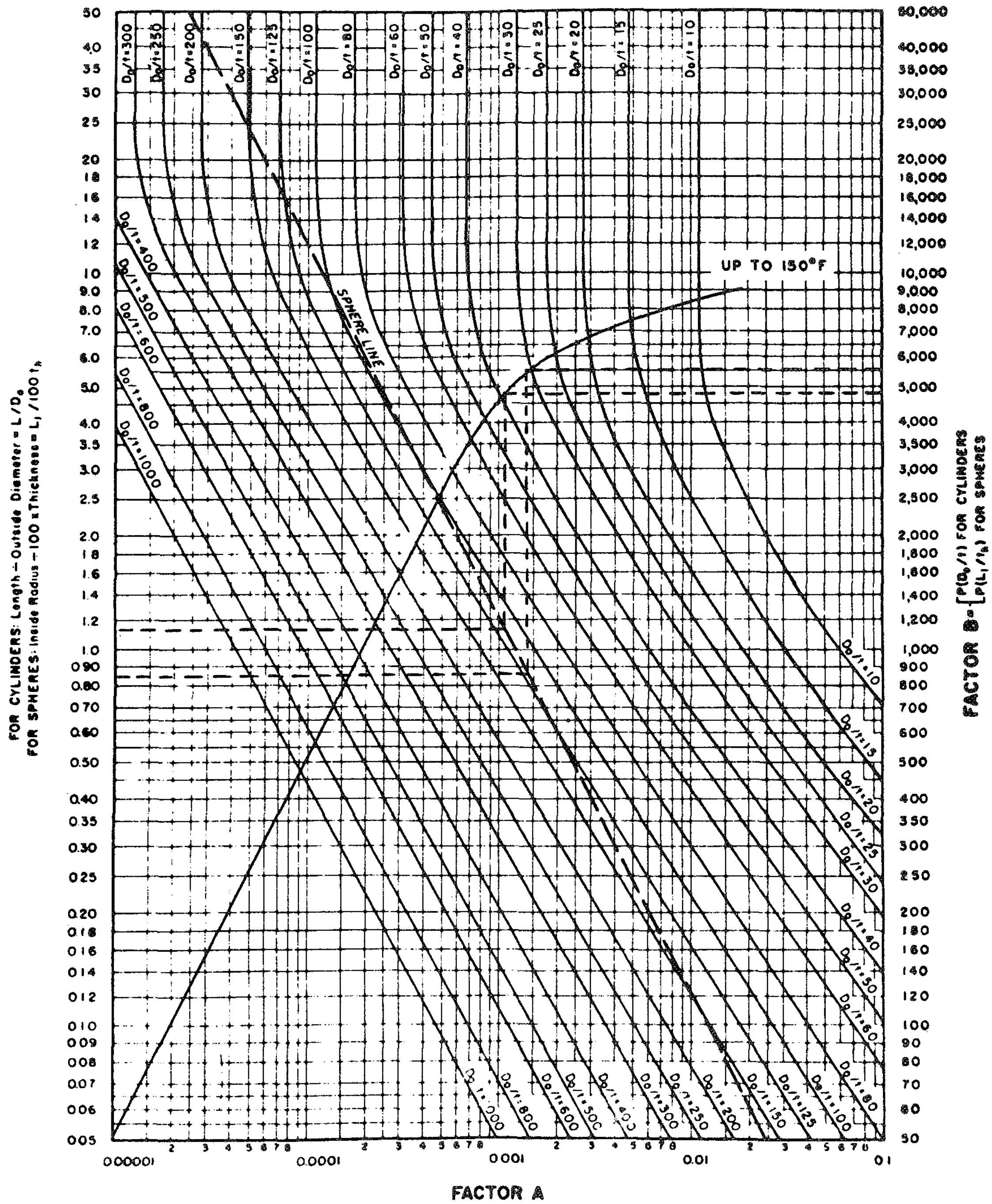

FIG. UNF-28.23 CHART FOR DETERMINING SHELL THICKNESS OF CYLINDRICAL AND SPHERICAL VESSELS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF ALUMINUM ALLOY 5083 (O or H-113 Temper) (For material having a Yield Strength not less than 18,000 psi)

ESIGN CALCULATIONS	COMPUTED BY ??E	REFERENCE DWGS.·
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 5 OF 46
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USE OF FIGURE UNF - 28.23 (SHEET #5)

LET $t = 0.25$ IN.

$$\frac{L}{D_o} = \frac{34}{30} = 1.13$$

$$\frac{D_o}{t} = \frac{30}{0.25} = 120$$

$$\beta = 4800$$

P_a = MAXIMUM ALLOWABLE WORKING PRESSURE

$$= \frac{\beta}{D_o/t} = \frac{4800}{120} = 40 \text{ PSI}$$

A SHELL THICKNESS OF 0.25 IN. IS THEREFORE WELL CAPABLE OF MEETING THE DESIGN REQUIREMENT OF 15 PSI EXTERNAL LOADING.

HEAD WALL THICKNESS

DESIGN DATA:

REFER TO DATA FOR SHELL WALL

FOR INTERNAL PRESSURE (UG-27, Pg. 11)

$$P = \frac{2SEt}{R + 0.2t}$$

WHEN $P \leq 0.665\,SE$

SUBSTITUTING:

$$P = \frac{2(10{,}000)(.7)(0.175)}{15 + 0.2(0.175)} = 163 \text{ PSI}$$

WHERE $t = 0.175$ IS MINIMUM HEAD TKS.

FOR EXTERNAL PRESSURE

THE PROCEDURE AS OUTLINED IN PARAGRAPH UG-28 (d) WILL BE FOLLOWED.

NOTATION

t_h = MINIMUM SPHERICAL SHELL TK'S

L_1 = SPHERICAL SHELL RADIUS
= 15 IN.

P_a = MAXIMUM ALLOWABLE PRESSURE

$$= \frac{B}{L_1/t_h}$$

SOLVING FOR THE PARAMETERS REQUIRED OF FIGURE UNF-28.23 (SHEET #5),

$$t_h = 0.175$$

$$\frac{L_1}{100 t_h} = \frac{15}{100(.175)} = 0.857$$

$$B = 5500$$

$$P_a = \frac{B}{L_1/t_h} = \frac{5500}{15/.175} = 64 \text{ PSI}$$

$\therefore$ A HEAD THICKNESS OF 0.080 IS CAPABLE OF MEETING BOTH THE EXTERNAL PRESSURE AND INTERNAL PRESSURE DESIGN REQUIREMENTS.

WEIGHT OF TANK

TOTAL MATERIAL VOLUME = VOLUME OF ONE SPHERE (30.00" O.D., 29.60" I.D.)* + VOLUME OF ONE CYLINDER (30.00" O.D., 29.50" I.D.)

$$VOL = \frac{1}{6}(3.14)\left[30.00^3 - 29.60^3\right] + \frac{3.14(24)}{4}\left[30.00^2 - 29.50^2\right]$$

$$= 558 + 561$$

$$= 1119 \text{ IN}^3$$

DENSITY OF 5083 AL. = 0.097 $LB/IN.^3$

1. TANK WEIGHT = 0.097(1119)
   = 109 LBS.

AREA COVERED BY INSULATION = AREA OF HEMISPHERE + AREA OF CYLINDER

$$A = \frac{\pi}{2}(30)^2 + \pi(30)(70) = 8011 \text{ IN}^2$$

$$= 55.6 \text{ FT.}^2$$

DENSITY OF INSULATION = 0.3 $LB./FT^2$

2. WEIGHT OF INSULATION = 55.6(0.3) = 17 LBS.

3. WEIGHT OF 6 IN. TRANSITION JOINT = 45 LBS.
   (Refer To "Tube-Turn" Engineering Data Sheet)

* ASSUME AN AVERAGE SPUN HEAD THICKNESS OF 0.200 IN.

DESIGN CALCULATIONS	COMPUTED BY: KKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 8 OF 46
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VOLUME OF FOAM INSULATION

$$\text{VOLUME OF FOAM} = \underset{\substack{(29.65''\text{ I.D.})\\(28''\text{ LONG})}}{\text{VOLUME OF CYLINDER}} - \underset{(29.84''\text{ I.D.})}{\text{VOLUME OF HEMISPHERE}}$$

$$= \frac{3.14(29.65)^2(28)}{4(1728)} - \frac{3.14(29.84)^3}{12(1728)}$$

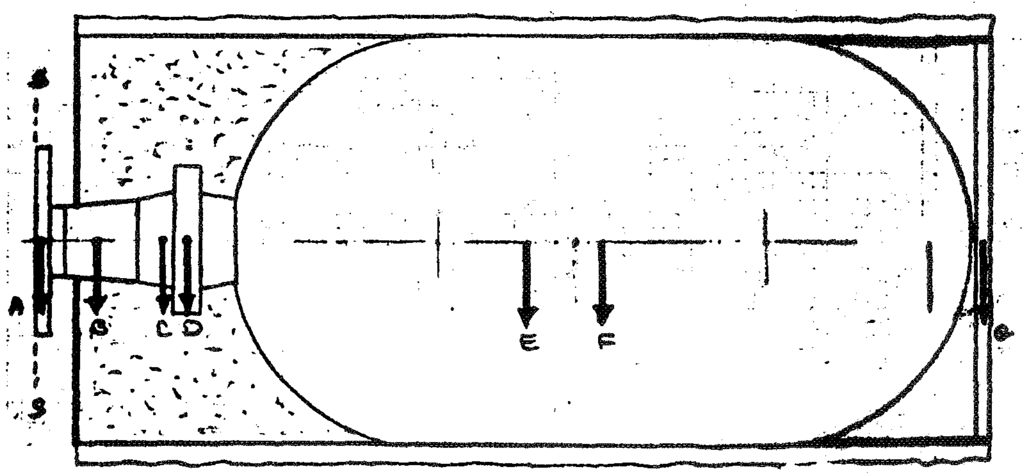
$$= 11.18 - 4.02$$

$$= 7.16\ FT^3$$

DENSITY OF FOAM ≈ 2.0 LBS/FT³

4. WEIGHT OF FOAM = 7.16 (2.0) ≈ 14.3 LBS

TOTAL WEIGHTS ACTING ON TANK

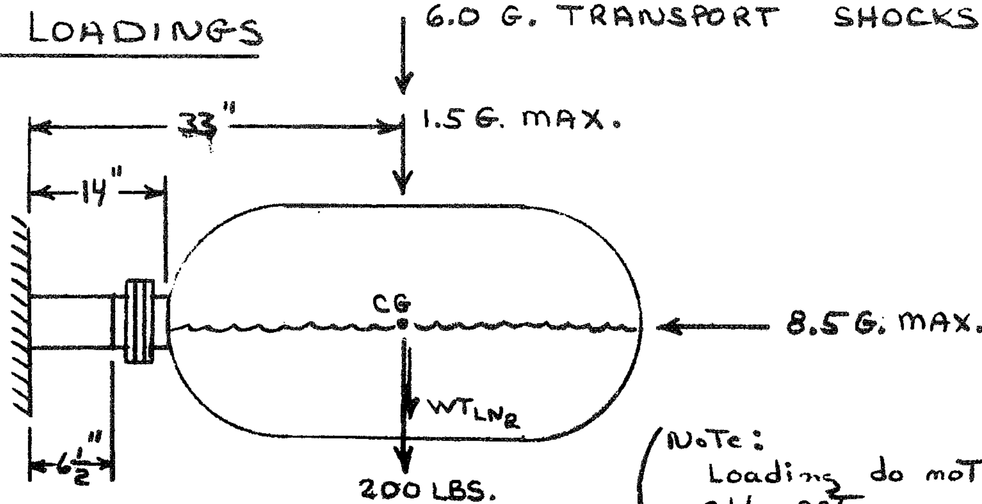


SUMMARY OF WEIGHTS

ITEM	DESCRIPTION	WEIGHT	DISTANCE FROM FLANGE SURFACE (S-S)
A	SUPPORT FLANGE	50 Lbs.	0.5 ins.
B	NECK TUBE REDUCER	5	4.5
C	URETHANE FOAM	14	9.0
D	TRANSITION JOINT	45	11.0
E	INSULATION	17	36.0
F	TANK	109	42.0
G	SKIRT	10	69.0

TOTAL = 250 LBS

DYNAMIC LOADINGS



(NOTE: Loadings do not all act simultaneously)

MAXIMUM LONGITUDINAL LOAD: 8.5 G. (NECK TUBE COMPRESSION)

$$P = 200(8.5) = 1700 \text{ LBS.}$$

MAXIMUM LATERIAL LOAD: 1.5 G. (NECK TUBE SHEAR)

$$P = 200(1.5) = 300 \text{ LBS.}$$

MAXIMUM MOMENT: 1.5 G. (NECK TUBE BENDING)

$$m = 200(1.5)(33) = 9900 \text{ IN.-LBS.}$$

NECK TUBE WILL BE CONSTRUCTED OF TYPE 304 STAINLESS STEEL (Seamless Tubing)

ALLOWABLE STRESS = 18,750 PSI.

(TAKEN FROM TABLE UHA-23 SPECIFICATION NO. SA-376)

DESIGN CALCULATIONS	COMPUTED BY: LKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 10 OF 46
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NECK TUBE WALL THICKNESS:

THE FOLLOWING LOADING CONDITIONS WILL BE CONSIDERED IN SIZING THE NECK TUBE WALL THICKNESS:

1. INTERNAL PRESSURE
2. EXTERNAL PRESSURE
3. COMPRESSIVE STRESS RESULTING FROM LONGITUDNAL COMPRESSION PLUS LATERIAL BENDING. (COMPARED WITH CRITICAL BUCKLING STRESS)
4. SHEAR AND COMPRESSIVE STRESS RESULTING FROM TANK HALF FILLED WITH LN_2 WHILE IN HORIZONTAL ATTITUDE
5. TENSILE STRESS WITH TANK IN VERTICAL POSITION AND FILLED WITH LN_2 (INTERNAL PRESSURE EFFECTS INCLUDED)
6. FLEXURE STRESS CAUSED BY TRANSPORT LATERIAL SHOCK LOADINGS
7. NATURAL FREQUENCY

NOTE:

ALTHOUGH A 6" TO 5" REDUCER IS USED IN THE NECK TUBE, A CONSTANT 5" DIA. OVER THE ENTIRE LENGTH WILL BE ASSUMED FOR COMPUTATIONS.

ITEM 1. DESIGN FOR INTERNAL PRESSURE (REF. UG-27, Pg. 11)

CIRCUMFERENTIAL STRESS (LONGITUDNAL JOINTS)

$$t = \frac{PR}{SE - 0.6P}$$

DESIGN CALCULATIONS	COMPUTED BY XKE	REFERENCE DWGS.:
	DATE	
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MATERIAL = TYPE 304 STAINLESS STEEL

WHERE t = MINIMUM REQUIRED THICKNESS
P = MAXIMUM DESIGN PRESSURE
= 75 PSI.
R = INSIDE RADIUS
= 2.5 IN.
S = MAXIMUM ALLOWABLE STRESS
= 18,750 PSI.
E = JOINT EFFICIENCY
= 1.0 (ASSUMING SEAMLESS TUBING)

SUBSTITUTING

$$t = \frac{75(2.5)}{18{,}750(1) - 0.6(75)} = 0.010 \text{ IN.}$$

ITEM 2. DESIGN FOR EXTERNAL PRESSURE
(REF. UG-28, PG. 11)

REFER TO FIGURE UHA-28.1 (SHEET 13) FOR DETERMINATION OF NECK TUBE THICKNESS UNDER EXTERNAL LOADING

WHERE t = MINIMUM REQ'D NECK TUBE THICKNESS
L = DESIGN LENGTH OF NECK TUBE
= 9 IN.
D_o = OUTSIDE NECK TUBE DIAMETER
= 5.56 IN.
P_a = MAXIMUM EXTERNAL DESIGN PRESSURE
= 15 PSI.

USE OF FIGURE UHA-28.1

USING t = 0.258 IN.

$$\frac{L}{D_o} = \frac{9}{5.56} = 1.618$$

$$\frac{D_o}{t} = \frac{5.56}{.258} = 21.55$$

DESIGN CALCULATIONS	COMPUTED BY: XKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D APP'V'D	SHEET NO. 12 OF 46

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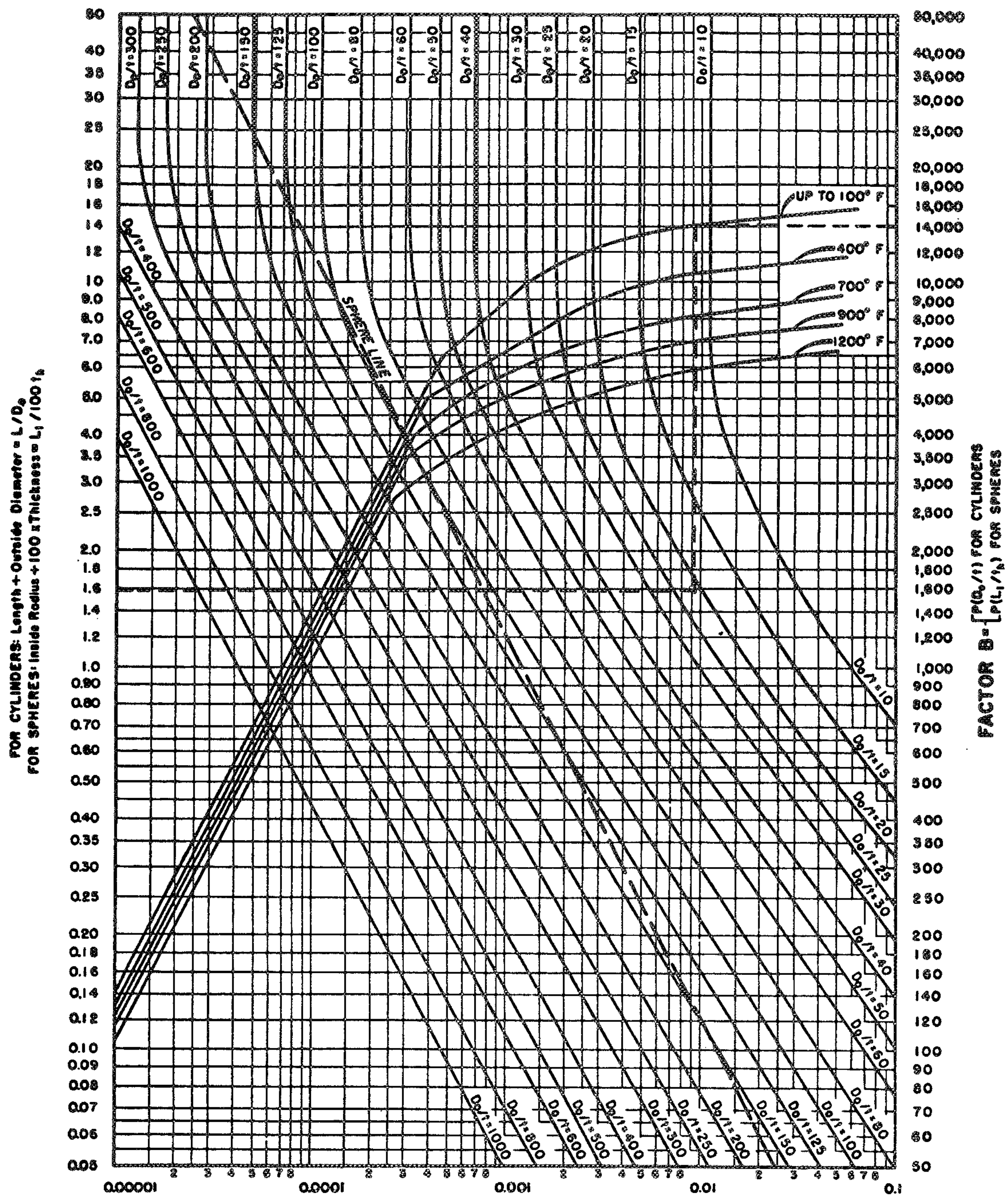

FIG. UHA-28.1 CHART FOR DETERMINING SHELL THICKNESS OF CYLINDRICAL AND SPHERICAL VESSELS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF AUSTENITIC STEEL (18 Cr-8 Ni, Type 304)

_ESIGN CALCULATIONS	COMPUTED BY YKE	REFERENCE DWGS.·
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 13 OF 46
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$B = 14,000$

$$P_a = \frac{B}{D_o/t} = \frac{14,000}{21.55} = 650 \text{ psi.}$$

$\therefore$ A NECK TUBE THICKNESS OF 0.258 IN. IS SUFFICIENT TO HANDLE THE EXTERNAL PRESSURE LOADING.

ITEM 3. COMPRESSIVE BUCKLING

FROM SHEET #10

MAXIMUM COMPRESSIVE LOAD = 1700 LBS.
MAXIMUM BENDING MOMENT = 9900 IN.-LBS.

$\therefore$ MAXIMUM COMPRESSIVE STRESS $= \dfrac{P}{A} + \dfrac{Mc}{I}$

WHERE

$$A = \frac{3.14}{4}\left[(5.56)^2 - (5.04)^2\right] = 4.327$$

(ASSUMING A 0.258 IN. WALL TKS.)

$$I = \frac{3.14}{64}\left[(5.56)^4 - (5.04)^4\right] = 15.229$$

SUBSTITUTING

$$\sigma_{MAX.} = \frac{1700}{4.327} + \frac{9900\,(2.78)}{15.229} = 2200 \text{ psi}$$

THIS MAXIMUM STRESS WILL NEXT BE COMPARED WITH THE CRITICAL COMPRESSIVE BUCKLING STRESS.

REFERENCE FOR CRITICAL BUCKLING STRESS

FORMULAS FOR STRESS AND STRAIN, RAYMOND J. ROARK, McGRAW HILL, FOURTH EDITION
(PAGE 352, CASES # 25 & 26)

CONSIDERING FIRST THE INFLUENCE OF LONGITUDINAL LOADING UPON A THIN WALLED TUBE,

$$s' = \frac{1}{\sqrt{3}}\left(\frac{Et}{r\sqrt{1-\nu^2}}\right)$$

WHERE s' = CRITICAL COMPRESSIVE STRESS
$E = 30 \times 10^6$ PSI
$t = 0.256$ IN.
$r = 2.78$
$\nu = 0.30$

$$s' = \frac{1}{\sqrt{3}}\left[\frac{30 \times 10^6 (.258)}{2.78\sqrt{.91}}\right] = 1.68 \times 10^6 \text{ PSI}$$

CONSIDERING NEXT THE INFLUENCE OF TRANSVERSE BENDING MOMENT UPON A THIN WALLED TUBE,

$$m' = K\left(\frac{E}{1-\nu^2}\right) r t^2$$

WHERE m' = CRITICAL BENDING MOMENT
K = CONSTANT (0.72 MINIMUM)

$$m' = .72\left(\frac{30 \times 10^6}{0.91}\right)(2.78)(.258)^2 = 4.39 \times 10^6 \text{ IN-LBS}$$

THE ABOVE COMPUTED CRITICAL VALUES EXCEED THE DESIGN REQUIREMENTS GIVEN ON SHEET # 14 AND FURTHER COMPUTATIONS INVOLVING THE SUPERPOSITION OF AXIAL AND BENDING LOADS TO ARRIVE AT A COMBINED CRITICAL WERE NOT DEVELOPED.

DESIGN CALCULATIONS	COMPUTED BY: LKE	REFERENCE DWGS.:
	DATE:	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D:	SHEET NO 15 OF 46
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ITEM 4. SHEAR AND COMPRESSIVE STRESS DUE TO LN_2 LOADING

THE TANK WILL BE IN A HORIZONTAL ATTITUDE AND COULD BE FILLED TO HALF OF ITS CAPACITY DURING COOLDOWN PRIOR TO VIBRATION TESTING

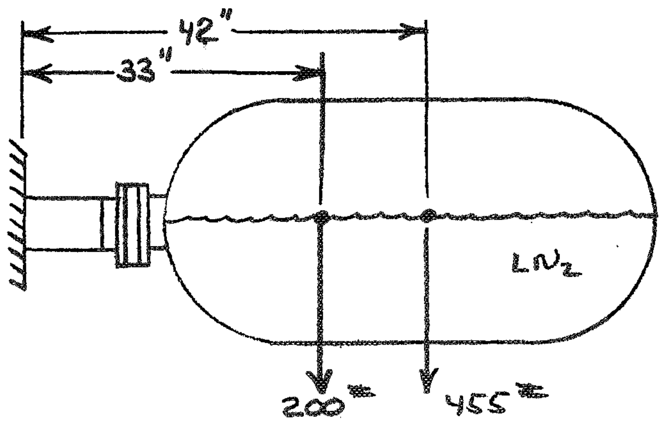
TOTAL TANK VOLUME = VOLUME OF SPHERE (30 IN. DIA.) + VOLUME OF CYLINDER (30 IN. DIA. 26 IN. LONG)

$$= \frac{3.14}{6}(30)^3 + \frac{3.14}{4}(24)(30)^2 = 31086 \text{ IN.}^2$$

$$= 18 \text{ CU. FT.}$$

DENSITY OF LIQUID NITROGEN = $50.46 \frac{LB}{FT^3}$

WEIGHT OF LN_2 WITH TANK ONE HALF FILLED = 455 LBS



EQUIVALENT LOAD OF 655 LBS ACTING AT 39 INCHES

DESIGN CALCULATIONS	COMPUTED BY YKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 16 OF 46
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IM L-174-4

SHEAR STRESS DUE TO BENDING $= \frac{P}{2A} = \frac{655}{\frac{3.14}{4}(5.563^2 - 5.05^2)(2)}$

$= 306$ PSI (NEGLIGIBLE)

COMPRESSIVE STRESS DUE TO BENDING $= \frac{Mc}{I} = \frac{39(655)(2.78)}{\frac{3.14}{64}\left[(5.56)^4 - (5.05)^4\right]}$

$= 4741$ PSI

THIS STRESS LEVEL IS BELOW THE MATERIAL YIELD POINT AND REPRESENTS A SAFE DESIGN CONDITION.

ITEM 5 TENSILE STRESS WHEN TANK IS IN VERTICAL POSITION AND FILLED WITH LIQUID NITROGEN AT 75 PSI

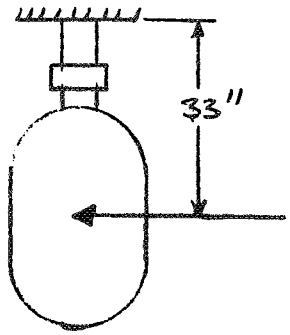
WT. OF TANK PLUS LN_2 = 200 + 910 = 1110 LBS

$$\sigma_{MAX.\ TENSILE} = \frac{P}{A} + \frac{PR}{2t}$$

$$= \frac{1100}{\frac{3.14}{4}(5.56^2 - 5.05^2)} + \frac{75(2.78)}{2(.258)}$$

$= 663$ PSI.

ITEM 6 FLEXURE STRESS CAUSED BY 6G TRANSPORT LATERIAL SHOCK



TANK WT. = 200 LBS

6 G. LOADING

NOTE
AN AMPLIFICATION FACTOR OF 1.2 WILL BE TAKEN.

MAXIMUM BENDING FORCE

$$m = 6(200)(1.2) = 1440 \text{ LBS.}$$

COMPRESSIVE STRESS DUE TO BENDING IN STEEL

$$\sigma = \frac{mc}{I}$$

$$= \frac{1440(33)(2.78)}{\frac{3.14}{64}\left[(5.56)^4 - (5.05)^4\right]}$$

$$= 8820 \text{ PSI}$$

SHEAR STRESS DUE TO BENDING $= \dfrac{P}{2A}$

$$= \frac{1440}{\frac{3.14}{4}\left[(5.56)^2 - (5.05)^2\right](2)}$$

$$= 678 \text{ PSI}$$

COMPRESSIVE STRESS DUE TO BENDING IN THE ALUMINUM END OF THE TRANSITION JOINT

(SEE SHEET 15 FOR CRITICAL STRESS COMPUTATIONS)

DESIGN CALCULATIONS	COMPUTED BY ZKE	REFERENCE DWGS.·
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 18 OF 46
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174-4

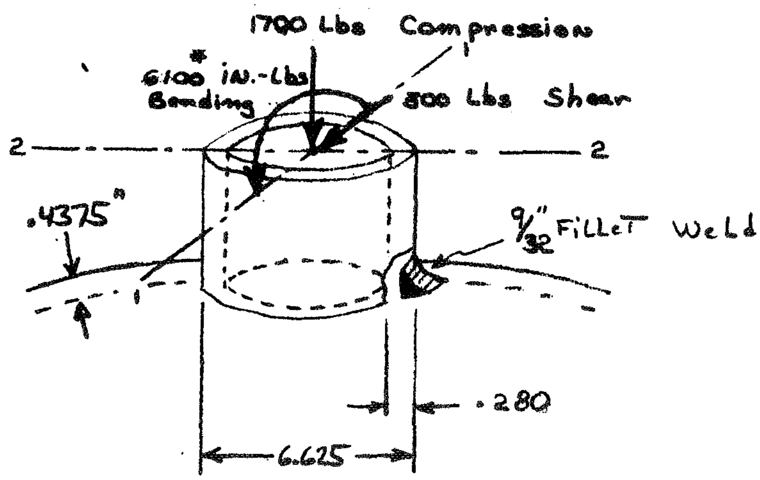
$$\tau = \frac{mc}{I} = \frac{1440\,(28.)\,(3.313)}{\frac{3.14}{64}\left[(6.625)^4 - (6.06)^4\right]}$$

$$= 4700 \text{ psi}$$

THE STRESS LEVELS FOR THIS NECK TUBE LOADING CONDITION ARE BELOW THE MATERIAL YIELD POINTS AND REPRESENT SAFE DESIGN CONDITIONS

STRESS LEVELS AT WELDED JOINTS

WELDED ATTACHMENT OF NECK TUBE TO SPHERICAL SHELL (REFERENCE: WELDING RESEARCH COUNCIL BULLETIN NO. 107 – "LOCAL STRESSES IN SPHERICAL AND CYLINDRICAL SHELLS DUE TO EXTERNAL LOADINGS")



* $m = \left[17(22.5) + 109(28.5) + 10(55.5)\right](1.5) = 6066$ IN.-LBS

USE 6100 IN-LBS FOR DESIGN CALCULATIONS

DETERMINE THE STRESS CONCENTRATION FACTOR REQUIRED FOR COMPUTATION SHEET NO. 21

$$\frac{r_A}{T} = \frac{\text{FILLET RADIUS}}{\text{PLATE THICKNESS}}$$

$$= \frac{0.281}{0.280}$$

$$= 1.0$$

USING FIGURE B-2

$K_n = 1.5$

$K_b = 1.3$

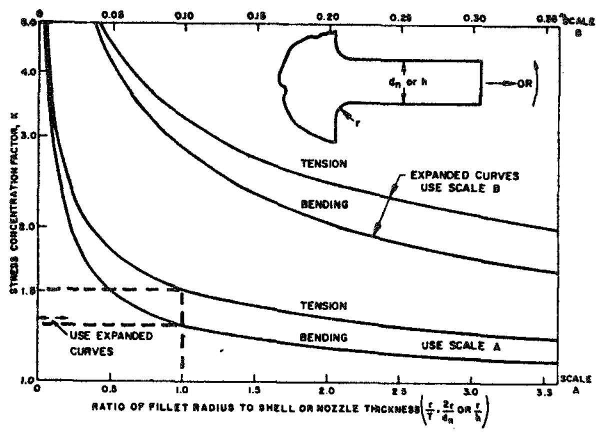


Fig. B-2—Stress concentration factors for D ≫ d

USE OF CURVES SP-1 TO SP-10

FIG. NO.	γ	ρ	$\frac{N_x T}{P}$	$\frac{N_Y T}{P}$	$\frac{m_x}{P}$	$\frac{m_y}{P}$
SP-2	5	1.00	.036	.055	.0275	.0088
SP-3	5	2.00	.43	.60	.011	.0098
SP-5	15	1.00	.20	.08	.035	.0105
SP-6	15	2.00	.023	.11	.019	.0145
INTERPOLATED	5	1.56	.257	.36	.0183	.0093
VALUES	15	1.56	.10	.096	.026	.0127
	11.3	1.56	.158	.194	.0232	.0114

ESIGN CALCULATIONS

UNION CARBIDE CORPORATION
LINDE DIVISION
TONAWANDA LABORATORIES
TONAWANDA, N Y 14152

COMPUTED BY LKE

DATE

CHK'D

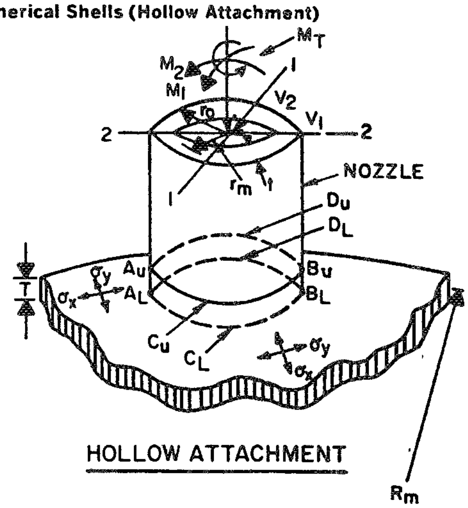
APP'V'D

REFERENCE DWGS.

SHEET NO 20 OF 46

D-

Table 3—Computation Sheet for Local Stresses in Spherical Shells (Hollow Attachment)



HOLLOW ATTACHMENT

1. Applied Loads*

Radial Load,	P	1700
Shear Load,	V_1	0
Shear Load,	V_2	300
Overturning Moment,	M_1	6100
Overturning Moment,	M_2	0
Torsional Moment,	M_T	0

2. Geometry

Vessel Thickness,	T	.4375
Vessel Mean Radius,	R_m	15
Nozzle Thickness,	t	.280
Nozzle Mean Radius,	r_m	3.17
Nozzle Outside Radius,	r_o	3.31

3. Geometric Parameters

γ	$\frac{r_m}{t}$	11.3
ρ	$\frac{T}{t}$	1.56
U	$\frac{r_o}{\sqrt{R_m T}}$	1.29

4. Stress Concentration Factors

due to:

membrane load, K_n 1.5

bending load, K_b 1.3

NOTE: Enter all force values in accordance with sign convention

From Fig.	Read curves for	Compute absolute values of stress and enter result →	Au	AL	Bu	BL	Cu	CL	Du	DL
			STRESSES – if load is opposite that shown, reverse signs shown							
SP–1 to 10	$\frac{N_xT}{P}$ .158	$K_n\left(\frac{N_xT}{P}\right)\cdot\frac{P}{T^2}$	-2105	-2105	-2105	-2105	-2105	-2105	-2105	-2105
↓	$\frac{M_x}{P}$ = .0232	$K_b\left(\frac{M_x}{P}\right)\cdot\frac{6P}{T^2}$	-1607	+1607	+1607	+1607	-1607	+1607	-1607	+1607
SM–1 to 10	$\frac{N_xT\sqrt{R_mT}}{M_1}$ .0356	$K_n\left(\frac{N_xT\sqrt{R_mT}}{M_1}\right)\cdot\frac{M_1}{T^2\sqrt{R_mT}}$					-665	-665	+665	+665
	$\frac{M_x\sqrt{R_mT}}{M_1}$ = .0323	$K_b\left(\frac{M_x\sqrt{R_mT}}{M_1}\right)\cdot\frac{6M_1}{T^2\sqrt{R_mT}}$					-3136	+3136	+3136	-3136
	$\frac{N_xT\sqrt{R_mT}}{M_2}$ .0356	$K_n\left(\frac{N_xT\sqrt{R_mT}}{M_2}\right)\cdot\frac{M_2}{T^2\sqrt{R_mT}}$	-0	-0	+0	+0				
↓	$\frac{M_x\sqrt{R_mT}}{M_2}$ .0323	$K_b\left(\frac{M_x\sqrt{R_mT}}{M_2}\right)\cdot\frac{6M_2}{T^2\sqrt{R_mT}}$	-0	+0	+0	-0				
Add algebraically for summation of σ_x,			-3712	-498	-498	-498	-7500	+1970	+89	-2969
SP–1 to 10	$\frac{N_yT}{P}$ = .194	$K_n\left(\frac{N_yT}{P}\right)\cdot\frac{P}{T^2}$ =	-2585	-2585	-2585	-2585	-2585	-2585	-2585	-2585
↓	$\frac{M_y}{P}$ .0114	$K_b\left(\frac{M_y}{P}\right)\cdot\frac{6P}{T^2}$	-790	+790	790	+790	-790	+790	-790	+790
SM–1 to 10	$\frac{N_yT\sqrt{R_mT}}{M_1}$ .082	$K_n\left(\frac{N_yT\sqrt{R_mT}}{M_1}\right)\cdot\frac{M_1}{T^2\sqrt{R_mT}}$					-1531	-1531	+1531	+1531
	$\frac{M_y\sqrt{R_mT}}{M_1}$ .0192	$K_b\left(\frac{M_y\sqrt{R_mT}}{M_1}\right)\cdot\frac{6M_1}{T^2\sqrt{R_mT}}$					-1864	+1864	-1864	1864
	$\frac{N_yT\sqrt{R_mT}}{M_2}$ .082	$K_n\left(\frac{N_yT\sqrt{R_mT}}{M_2}\right)\cdot\frac{M_2}{T^2\sqrt{R_mT}}$	-0	-0	+0	+0				
↓	$\frac{M_y\sqrt{R_mT}}{M_2}$ .0192	$K_b\left(\frac{M_y\sqrt{R_mT}}{M_2}\right)\cdot\frac{6M_2}{T^2\sqrt{R_mT}}$	-0	+0	+0	-0				
Add algebraically for summation of σ_y,			-3375	-1795	-3375	-1795	-6770	-1462	+20	-2128
Shear stress due to load, V_1		$\tau_1 = \frac{V_1}{\pi r_o T}$ =					-0	-0	+0	+0
Shear stress due to load, V_2		$\tau_2 = \frac{V_2}{\pi r_o T}$ =	66	66	66	66				
Shear stress due to torsion, M_T		$\tau_3 = \frac{M_T}{2\pi r_o^2 T}$ =	0	0	0	0	0	0	0	0
Add algebraically for summation of τ,										
COMBINED STRESS INTENSITY, S										
When σ_x & σ_y have like signs $S = \frac{1}{2}\left[\sigma_x + \sigma_y \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}\right]$ When $\tau = 0$, S = largest of σ_x, σ_y, or $\lvert\sigma_x - \sigma_y\rvert$ When σ_x & σ_y have unlike signs $S = \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}$							-7500			

USE OF CURVES SM-1 TO SM-10

FIG. NO.	γ	ρ	$\frac{N_x T \sqrt{R_m T}}{M}$	$\frac{N_Y T \sqrt{R_m T}}{M}$	$\frac{M_x \sqrt{R_m T}}{M}$	$\frac{M_Y \sqrt{R_m T}}{M}$
SM-2	5	1.00	.038	.056	.038	.016
SM-3	5	2.00	.070	.040	.016	.016
SM-5	15	1.00	.023	.074	.047	.017
SM-6	15	2.00	.024	.125	.027	.024
INTERPOLATED	5	1.56	.056	.047	.026	.016
VALUES	15	1.56	.0236	.103	.036	.021
	11.3	1.56	.0356	.082	.0323	.0192

THE RESULTS INDICATE A MAXIMUM STRESS BELOW 7500 PSI AT THE WELDED CONNECTION OF THE TRANSITION JOINT TO THE HEMISPHERICAL HEAD FOR THE SPECIFIED LOADING CONDITION. THIS IS WITHIN THE YIELD STRESS OF 5083-O ALUMINUM (21,000 PSI) AND ACCEPTABLE.

A SECOND DESIGN CONDITION OF THE WELDED JOINT BETWEEN THE NECK TUBE AND HEMISPHERICAL HEAD INVOLVES THE LOADING ENCOUNTERED DURING TRANSPORT.

$$\text{BENDING MOMENT} = \left[17(22.5) + 109(28.5) + 10(56.5)\right] 6\,(1.2)$$

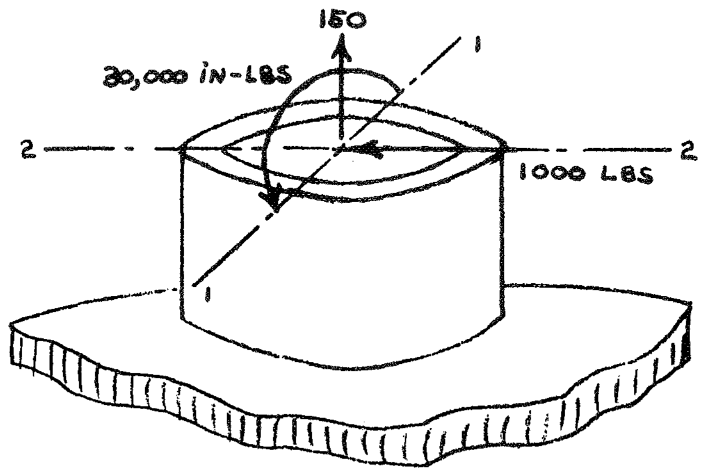
$$= 29{,}100 \text{ IN-LBS}$$

(USE 30,000 IN-LBS FOR DESIGN CALCULATIONS)

A CALCULATING PROCEDURE SIMILAR TO THAT OF THE PREVIOUS CASE WILL BE EMPLOYED.

$$\text{SHEARING FORCE} \approx (17 + 109 + 10)(6)(1.2) = 980 \text{ LBS}$$

(USE 1000 LBS FOR DESIGN CALCULATIONS)

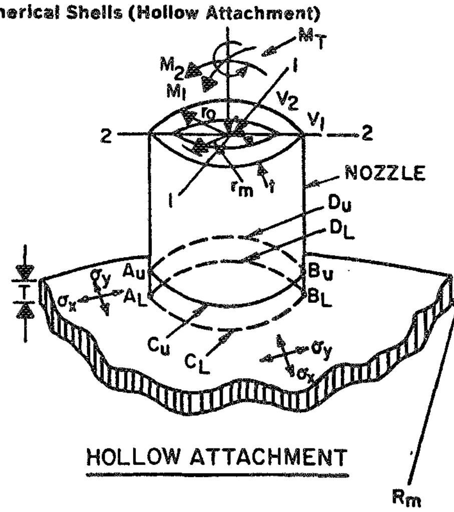


THE RESULTS INDICATE A STRESS LEVEL OF APPROXIMATELY 19,000 PSI. THIS VALUE IS CLOSE TO THE MATERIAL YIELD POINT BUT ACCEPTABLE. THIS LOADING CONDITION REPRESENTS AN EXTREME WHICH IS NOT LIKELY TO OCCUR FREQUENTLY DURING TRANSPORT.

DESIGN CALCULATIONS	COMPUTED BY LKE	REFERENCE DWGS.·
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 23 OF 46
	APP'V'D	D-

I 9M -174-4

Table 3—Computation Sheet for Local Stresses in Spherical Shells (Hollow Attachment)



1. Applied Loads*
   - Radial Load, P = 150
   - Shear Load, V_1 = 1000
   - Shear Load, V_2 = 0
   - Overturning Moment, M_1 = 30,000
   - Overturning Moment, M_2 = 0
   - Torsional Moment, M_T = 0

2. Geometry
   - Vessel Thickness, T = .4375
   - Vessel Mean Radius, Rm = 15
   - Nozzle Thickness, t = .280
   - Nozzle Mean Radius, r_m = 3.17
   - Nozzle Outside Radius, r_o = 3.31

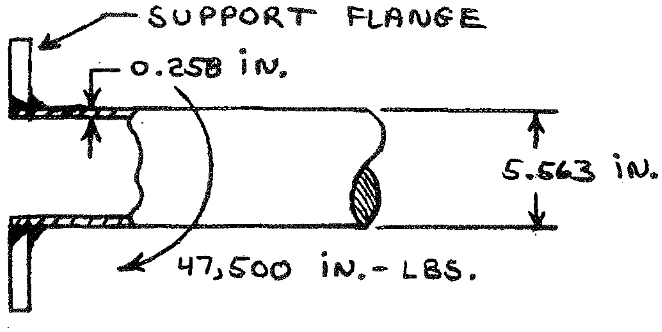
3. Geometric Parameters
   - $\gamma = \frac{r_m}{t}$ = 11.3
   - $\rho = \frac{T}{t}$ = 1.56
   - $U = \frac{r_o}{\sqrt{R_m T}}$ = 1.29

4. Stress Concentration Factors due to:
   - membrane load, K_n = 1.5
   - bending load, K_b = 1.3

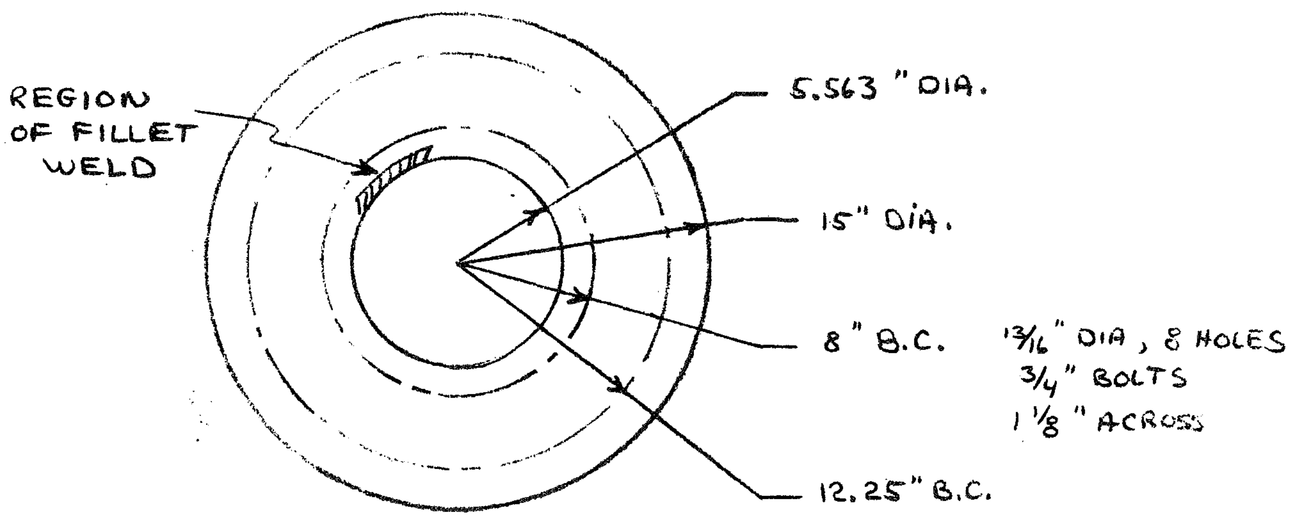
NOTE: Enter all force values in accordance with sign convention

From Fig.	Read curves for	Compute absolute values of stress and enter result →	Au	AL	Bu	BL	Cu	CL	Du	DL
			STRESSES – if load is opposite that shown, reverse signs shown							
SP-1 to 10	$\frac{N_xT}{P}$ = .158	$K_n \left(\frac{N_xT}{P}\right) \cdot \frac{P}{T^2}$	+186	+ 186	+ 186	+ 186	+ 186	+ 186	+ 186	+186
↓	$\frac{M_x}{P}$ = .0232	$K_b \left(\frac{M_x}{P}\right) \cdot \frac{6P}{T^2}$ =	+142	– 142	– 142	– 142	+ 142	– 142	+142	– 142
SM-1 to 10	$\frac{N_xT\sqrt{R_mT}}{M_1}$ = .0356	$K_n \left(\frac{N_xT\sqrt{R_mT}}{M_1}\right) \cdot \frac{M_1}{T^2\sqrt{R_mT}}$					–3269	–3269	+3269	+3269
↓	$\frac{M_x\sqrt{R_mT}}{M_1}$ = .0323	$K_b \left(\frac{M_x\sqrt{R_mT}}{M_1}\right) \cdot \frac{6M_1}{T^2\sqrt{R_mT}}$					–15,400	+15,400	+15400	–15400
	$\frac{N_xT\sqrt{R_mT}}{M_2}$	$K_n \left(\frac{N_xT\sqrt{R_mT}}{M_2}\right) \cdot \frac{M_2}{T^2\sqrt{R_mT}}$ =	– 0	– 0	+ 0	+ 0				
↓	$\frac{M_x\sqrt{R_mT}}{M_2}$	$K_b \left(\frac{M_x\sqrt{R_mT}}{M_2}\right) \cdot \frac{6M_2}{T^2\sqrt{R_mT}}$	– 0	+ 0	+ 0	– 0				
Add algebraically for summation of σ_x, =			+328	+44	+44	+ 44	–18300	+12175	+19000	–12100
SP-1 to 10	$\frac{N_yT}{P}$ = .194	$K_n \left(\frac{N_yT}{P}\right) \cdot \frac{P}{T^2}$ =	+228	+228	+228	+228	+228	+228	+228	+228
↓	$\frac{M_y}{P}$ = .0114	$K_b \left(\frac{M_y}{P}\right) \cdot \frac{6P}{T^2}$	+69	– 69	+69	– 69	+69	– 69	+69	– 69
SM-1 to 10	$\frac{N_yT\sqrt{R_mT}}{M_1}$ = .082	$K_n \left(\frac{N_yT\sqrt{R_mT}}{M_1}\right) \cdot \frac{M_1}{T^2\sqrt{R_mT}}$					–7530	– 7530	– 7530	+7530
↓	$\frac{M_y\sqrt{R_mT}}{M_1}$ = .0192	$K_b \left(\frac{M_y\sqrt{R_mT}}{M_1}\right) \cdot \frac{6M_1}{T^2\sqrt{R_mT}}$					–9170	+9170	– 9170	–9170
	$\frac{N_yT\sqrt{R_mT}}{M_2}$	$K_n \left(\frac{N_yT\sqrt{R_mT}}{M_2}\right) \cdot \frac{M_2}{T^2\sqrt{R_mT}}$	– 0	– 0	+ 0	+ 0				
↓	$\frac{M_y\sqrt{R_mT}}{M_2}$	$K_b \left(\frac{M_y\sqrt{R_mT}}{M_2}\right) \cdot \frac{6M_2}{T^2\sqrt{R_mT}}$ =	– 0	+ 0	+ 0	– 0				
Add algebraically for summation of σ_y, =			+297	–159	+297	–159	–16400	+1800	+17000	–1500
Shear stress due to load, V_1		$\tau_1 = \frac{V_1}{\pi r_o T}$ =					219	219	219	219
Shear stress due to load, V_2		$\tau_2 = \frac{V_2}{\pi r_o T}$ =	0	0	0	0				
Shear stress due to torsion, M_T		$\tau_3 = \frac{M_T}{2\pi r_o^2 T}$ =	0	+ 0	0	0	0	0	0	0
Add algebraically for summation of τ, =										
COMBINED STRESS INTENSITY, S										
When σ_x & σ_y have like signs: $S = \frac{1}{2}\left[\sigma_x + \sigma_y + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}\right]$ When τ = 0, S = largest of σ_x, σ_y, or $\lvert\sigma_x - \sigma_y\rvert$ When σ_x & σ_y have unlike signs: $S = \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}$									19000	

THE STRESS LEVEL AT THE STAINLESS STEEL WELDED JOINT BETWEEN THE NECK TUBE AND SUPPORT FLANGE WILL BE DETERMINED.



SUPPORT FLANGE LAYOUT



8.000
5.563

$2 \overline{)2.437}$ = 1.218"

$2 \overline{)1.125}$ = .563

APPROX. SPACE AVAILABLE FOR FILLET WELD:

1.218 − .563 = .655 IN.

DESIGN CALCULATIONS	COMPUTED BY: LKE	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 25 OF 46
	APP'V'D	

FORM L-174-4

STRESS AT THROAT OF FILLET WELD

$$S = \frac{5.66\,M}{\pi h D^2}$$ (REF: WELDING HANDBOOK, 3d ed., P.P. 1566-67, 1950)

ASSUME A 1/4" FILLET WELD

$$S = \frac{5.66(47{,}500)}{3.14(.25)(5.563)^2} = 11{,}060 \text{ PSI.}$$

SHEARING STRESS IN FUSION ZONE OF WELD

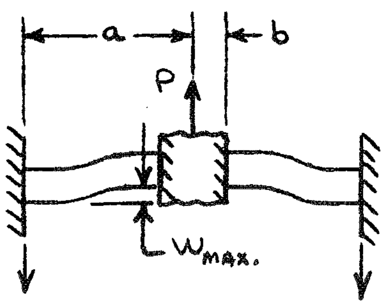
$$S_s = \frac{M}{h r^2 \pi}$$

$$= \frac{47{,}500}{.25\left(\frac{5.563}{2}\right)^2(3.14)}$$

$$= 7{,}820 \text{ PSI.}$$

MAXIMUM STRESS AND DEFLECTION OF A SOLID PLATE AS SHOWN BELOW

REF: STRENGTH OF MATERIALS, TEMOSHENKO, 3nd Ed., VAN NOSTRAND CO., 1956, PP 113-14



$$\sigma_{MAX} = \frac{k P}{h^2}$$

$$W_{MAX} = k_1 \frac{P a^2}{E h^3}$$

DESIGN CALCULATIONS	COMPUTED BY LKC	REFERENCE DWGS.:
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D	SHEET NO. 26 OF 46
	APP'V'D	

RM L-174-4

$k \,\&\, k_1$ = COEFFICIENT (TABLE 5, PG 114)

WHERE: h = PLATE THICKNESS (0.75 IN.)

P = TOTAL LOAD (1275 LBS)

E = MODULUS OF ELASTICITY (30,000,000 PSI)

$a/b = 6.125/2.78 = 2.2$

(ASSUME 2 FOR DESIGN)

$$\tau_{MAX} = \frac{.7 (1275)}{(.75)^2} = 1586 \text{ psi.}$$

$$w_{MAX} = 0.06 \left[\frac{1275 \cdot (6.125)^2}{30 \times 10^6 (.75)^3} \right] = 0.0003 \text{ in.}$$

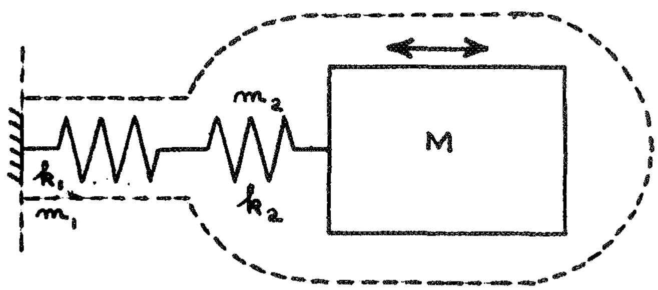
THESE FIGURES DO NOT INCLUDE THE PRESENCE OF BOLT HOLES WITHIN THE PLATE BUT DO SHOW A VERY LOW STRESS AND DEFLECTION LEVEL FOR THE SOLID PLATE.

ITEM 7 NATURAL FREQUENCY

REFERENCES: "MECHANICAL VIBRATIONS", DEN HARTOG, Mc GRAW HILL, 1956

"FORMULAS FOR STRESS AND STRAIN", RAYMOND J. ROARK, Mc GRAW HILL, 1965

THE TANK HAS BEEN MODELED AS A LARGE SPRING CONNECTED MASS FOR COMPUTATION PURPOSES.



* ASSUME $\frac{1}{3}$ OF HEMISPHERE MASS ACTS IN VIBRATION

FOR LONGITUDONAL VIBRATION

$$\omega_n = \sqrt{\frac{k}{(M + m/3)}}$$

WHERE
ω_n = NATURAL FREQUENCY
M = MASS OF TANK
m = MASS OF NECK TUBE PLUS EFFECTIVE MASS OF SHELL
k = SPRING CONSTANT

$$M = \frac{W}{g_c} = \frac{(17 + 109 + 10)\ LBS_m}{386\ \frac{LBS_m - IN}{LBS_f - Sec^2}} = 0.352\ \frac{LBS_f - Sec^2}{IN.}$$

$$m = m_1 + m_2 = \frac{(5 + 14 + 45 + 10^*)}{386} = 0.192$$

DESIGN CALCULATIONS	COMPUTED BY LXE	REFERENCE DWGS.
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 28 OF 46
	APP'V'D	D-

9N -174-4

k_1 = SPRING CONSTANT OF NECK TUBE

$$= \frac{P}{e} = \frac{AE}{\ell}$$

CONSIDER THE TUBE TO BE 5" NPS x 9" LG
CONSTRUCTED OF STEEL

THEN,

$$k_1 = \frac{\frac{3.14}{4}\left[(5.563)^2-(5.05)^2\right](30\times10^6)}{9}$$

$$= 14.25 \times 10^6 \text{ LBS/IN.}$$

k_2 = SPRING CONSTANT OF HEMISPHERE

CONSIDER LOAD CONCENTRATED ON CIRCULAR AREA OF RADIUS r_o. VERTICAL SUPPORTS WITH EDGES NEITHER HELD NOR FIXED.

$$k_2 = \frac{P}{y} = \frac{Et^2}{RA}$$

REF. ROARK, Pg 304, Case #20

WHERE A IS A NUMERICAL COEFF. DEPENDING UPON

$$\mu = \sqrt[4]{12(1-\nu^2)}\left(\frac{r_o}{\sqrt{R_2 t}}\right)$$

AND HAS VALUES AS TABULATED BELOW:

μ	0	.1	.2	.4	.6	.8	1.0	1.2	1.4
A	.424	.418	.410	.405	.381	.354	.330	.305	.280

SOLVING FOR μ

$$\mu = \sqrt[4]{12(1-.0625)}\left(\frac{5}{\sqrt{15(.4375)}}\right) = 3.57$$

THE EXTRAPOLATED VALUE OF A = 0.02

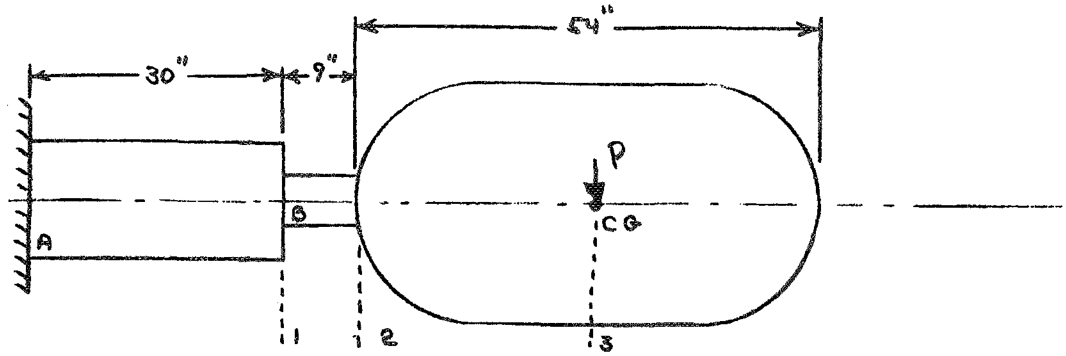
$$k_2 = \frac{30 \times 10^6 (.4375)^2}{15(.02)} = 19 \times 10^6 \ lbs/in.$$

$$k = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2}} = \frac{1}{\frac{1}{14} + \frac{1}{19}} = 8 \times 10^6 \ lbs/in$$

$$\omega_n = \sqrt{\frac{8 \times 10^6}{.352 + \frac{.192}{3}}} = 4400 \ rad/sec$$

$$= 700 \ cycles/sec$$

FOR TRANSVERSE VIBRATION



MATERIALS:

A. 14" O.D. x 0.180" TK. ALUMINUM (5083-O)

B. 5.583" O.D x 0.258" TK. STAINLESS STEEL

DEFLECTION OF STRUCTURE AT POINT (1)

a. DUE TO END LOAD

$$\delta_1 = \frac{P\ell^3}{3EI} = \frac{250\,(30)^3}{3(17 \times 10^6)\left(\frac{3.14}{64}\right)\left[(14)^4 - (13.64)^4\right]}$$

$\delta_1 = 0.000709$ IN.

b. DUE TO END MOMENT

$$\delta_1 = \frac{m\ell^2}{2EI} = \frac{250\,(27+9)(30)^2}{2(17 \times 10^6)\left(\frac{3.14}{64}\right)\left[(14)^4 - (13.64)^4\right]}$$

$\delta_1 = 0.001277$ IN.

BEAM ROTATION AT POINT (1)

a. DUE TO END LOAD

$$\theta_1 = \frac{W\ell^2}{2EI} = \frac{250\,(30)^2}{2\,(17 \times 10^6)\left(\frac{3.14}{64}\right)\left[(14)^4 - (13.64)^4\right]}$$

$\theta_1 = 0.000035$ rad

DESIGN CALCULATIONS | COMPUTED BY KKE | REFERENCE DWGS.· | DATE REV 2/71 | CHK'D | SHEET NO 31 OF 46 | APP'V'D | D-

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TONAWANDA, N Y 14152

FORM -174-6

b. DUE TO END MOMENT

$$\theta_1 = \frac{M\ell}{EI} = \frac{250(27+9)(30)}{17\times10^6\left(\frac{3.14}{64}\right)\left[(14)^4-(13.64)^4\right]}$$

$\theta_1 = 0.000085$ rad.

DEFLECTION OF STRUCTURE AT POINT (2) HOLDING POINT (1) FIXED

a. DUE TO END LOAD

$$\delta_2 = \frac{P\ell^3}{3EI} = \frac{250(9)^3}{3(30\times10^6)\left(\frac{3.14}{64}\right)\left[(5.56)^4-(5.05)^4\right]}$$

$\delta_2 = 0.000135$ IN.

b. DUE TO END MOMENT

$$\delta_2 = \frac{M\ell^2}{2EI} = \frac{250(27)(9)^2}{2(30\times10^6)\left(\frac{3.14}{64}\right)\left[(5.56)^4-(5.05)^4\right]}$$

$\delta_2 = 0.000608$ IN.

BEAM ROTATION AT POINT (2)

a. DUE TO END LOAD

$$\theta_2 = \frac{W\ell^2}{2EI} = \frac{250(9)^2}{2(30\times10^6)\left(\frac{3.14}{64}\right)\left[(5.56)^4-(5.05)^4\right]}$$

$\theta_2 = 0.000022$

b. DUE TO END MOMENT

$$\theta_2 = \frac{m\ell}{EI} = \frac{250(27)(9)}{30 \times 10^6 \left(\frac{3.14}{64}\right)\left[(5.56)^4 - (5.05)^4\right]}$$

$\theta_2 = 0.000135$ rad.

DEFLECTION AT CONTAINER CENTER OF GRAVITY

a. DUE TO END LOAD AND MOMENT LOAD

$$\delta_{CG} = 0.000709 + 0.001277 + 0.000135 + .000608$$

$$= 0.00273 \text{ IN.}$$

b. DUE TO ROTATION

$$\delta_{CG} = (9 + 27)(.000035 + .000085) + (27)(.000022 + .000135)$$

$$= 0.00855 \text{ IN.}$$

TOTAL DEFLECTION AT CENTER OF GRAVITY

$$\delta_{CG} = 0.00273 + .00855 = 0.0113 \text{ IN.}$$

$$k = \frac{P}{\delta} = \frac{250}{.0113} = 22124$$

$$\omega_n = \sqrt{\frac{k}{m}}$$

DESIGN CALCULATIONS	COMPUTED BY LKE	REFERENCE DWGS.-
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 33 OF 46
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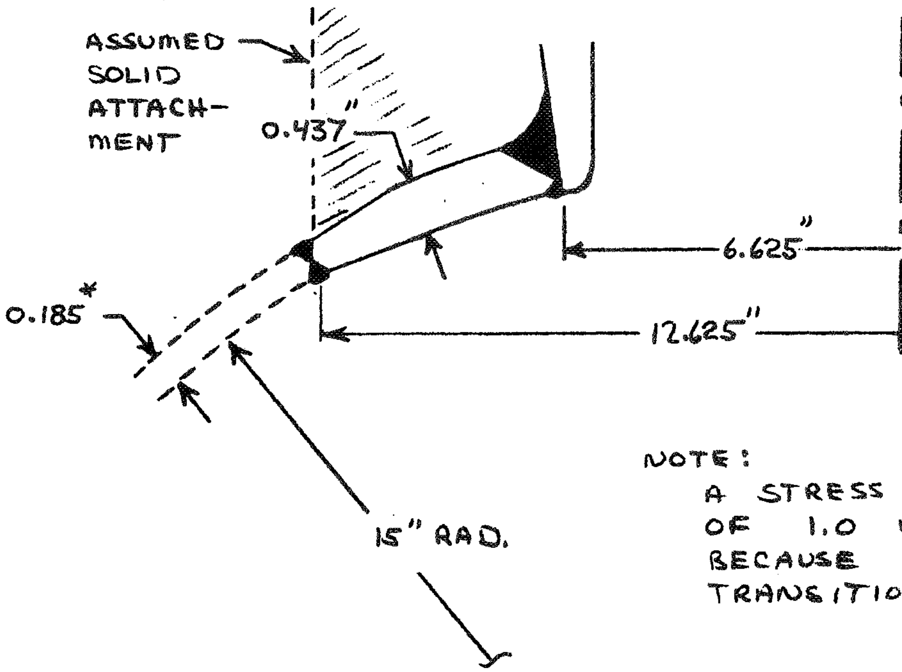
$$m = \frac{250 \text{ Lbs}}{386 \ \frac{\text{Lbs}_m - \text{IN}}{\text{Lbs}_f - \text{Sec}^2}} = 0.648 \ \frac{\text{Lbs}_f - \text{Sec}^2}{\text{IN.}}$$

$$\omega_n = \sqrt{\frac{22,124}{0.648}} = 185 \text{ rad/sec}$$

$$= 29 \text{ cycles/sec}$$

CHECK OF WELD JOINT BETWEEN SPHERICAL HEAD AND REINFORCEMENT RING

A SOLID ATTACHMENT WILL BE ASSUMED TO FACILITATE USING THE CALCULATING PROCEDURE AS OUTLINED IN THE "WELDING RESEARCH COUNCIL BULLETIN NO. 107".



NOTE:
A STRESS CONCENTRATION FACTOR OF 1.0 WILL BE USED BECAUSE OF THE GRADUAL TRANSITION IN THICKNESSES

SHEET #36 SUMMARIZES THE STRESS COMPUTATION. A STRESS LEVEL LESS THAN 17,000 PSI SHOULD EXIST MOMENTARILY AT THE SUBJECT WELD AS A RESULT OF TRANSVERSE SHOCK LOADS OF 6G'S. THIS IS ACCEPTABLE FOR SHORT DURATIONS.

* A CONSERVATIVE ESTIMATE BASED UPON THE ANTICIPATED VARIATION IN THICKNESS DUE TO THE SPINNING OPERATION ($t = 0.175 \rightarrow 0.250$)

DESIGN CALCULATIONS	COMPUTED BY: LKE	REFERENCE DWGS.:
	DATE: REV. 2/71	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N. Y. 14152	CHK'D:	SHEET NO. 35 OF 46
	APP'V'D:	

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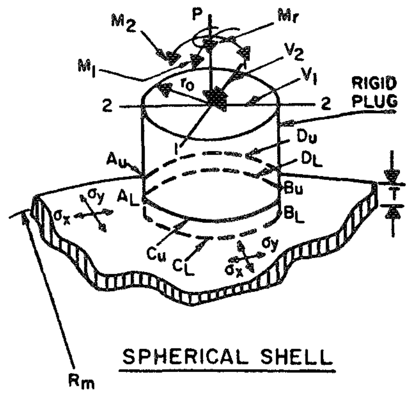
Table 2—Computation Sheet for Local Stresses in Spherical Shells (Solid Attachment)



1. Applied Loads*

Radial Load,	P	~0
Shear Load,	V_1	0
Shear Load,	V_2	0
Overturning Moment,	M_1	30,000
Overturning Moment,	M_2	0
Torsional Moment,	M_T	0

2. Geometry

Vessel Thickness,	T	0.185
Vessel Mean Radius,	Rm	15
Attachment Outside Radius,	r_o	6.313

3. Geometric Parameters

$U = \frac{r_o}{\sqrt{R_m T}} = 3.79$

$\sqrt{R_m T}$ 1.67

$T\sqrt{R_m T}$ .057

4. Stress Concentration Factors due to:

membrane load, Kn 1

bending load, Kb 1

*NOTE: Enter all force values in accordance with sign convention

Reference Fig. Nos.	Read curves for	Calculate absolute values of stress and enter result	Au	AL	Bu	BL	Cu	CL	Du	DL
		STRESSES – if load is opposite that shown, reverse signs shown								
SR – 2	$\frac{N_xT}{P}$	$K_n\left(\frac{N_xT}{P}\right)\cdot\frac{P}{T^2}$	0	0	- 0	0	- 0	0	- 0	- 0
↓	$\frac{M_x}{P}$	$K_b\left(\frac{M_x}{P}\right)\cdot\frac{6P}{T^2}$	- 0	0	0	0	0	+ 0	- 0	+ 0
SR – 3	$\frac{N_xT\sqrt{R_mT}}{M_1}$ .0076	$K_n\left(\frac{N_xT\sqrt{R_mT}}{M_1}\right)\cdot\frac{M_1}{T^2\sqrt{R_mT}}$					4000	-4000	+4000	+4000
	$\frac{M_x\sqrt{R_mT}}{M_1}$ .0042	$K_b\left(\frac{M_x\sqrt{R_mT}}{M_1}\right)\cdot\frac{6M_1}{T^2\sqrt{R_mT}}$					13200	-13200	+13200	-13200
	$\frac{N_xT\sqrt{R_mT}}{M_2}$	$K_n\left(\frac{N_xT\sqrt{R_mT}}{M_2}\right)\cdot\frac{M_2}{T^2\sqrt{R_mT}}$	0	0	0	0				
↓	$\frac{M_x\sqrt{R_mT}}{M_2}$	$K_b\left(\frac{M_x\sqrt{R_mT}}{M_2}\right)\cdot\frac{6M_2}{T^2\sqrt{R_mT}}$	0	0	- 0	0				
Add algebraically for summation of radial stresses, σ_x		σ_x =					-17200	+9200	+17200	-9200
SR – 2	$\frac{N_yT}{P}$	$K_n\left(\frac{N_yT}{P}\right)\cdot\frac{P}{T^2}$	0	- 0	0	- 0	- 0	- 0	- 0	- 0
↓	$\frac{M_y}{P}$	$K_b\left(\frac{M_y}{P}\right)\cdot\frac{6P}{T^2}$	0	0	0	+ 0	- 0	+ 0	+ 0	+ 0
SR – 3	$\frac{N_yT\sqrt{R_mT}}{M_1}$ .0022	$K_n\left(\frac{N_yT\sqrt{R_mT}}{M_1}\right)\cdot\frac{M_1}{T^2\sqrt{R_mT}}$					-1150	-1150	+1150	+1150
	$\frac{M_y\sqrt{R_mT}}{M_1}$ .0013	$K_b\left(\frac{M_y\sqrt{R_mT}}{M_1}\right)\cdot\frac{6M_1}{T^2\sqrt{R_mT}}$					-4100	+4100	+4100	-4100
	$\frac{N_yT\sqrt{R_mT}}{M_2}$	$K_n\left(\frac{N_yT\sqrt{R_mT}}{M_2}\right)\cdot\frac{M_2}{T^2\sqrt{R_mT}}$	0	0	0	0				
↓	$\frac{M_y\sqrt{R_mT}}{M_2}$	$K_b\left(\frac{M_y\sqrt{R_mT}}{M_2}\right)\cdot\frac{6M_2}{T^2\sqrt{R_mT}}$	0	0	0	0				
Add algebraically for summation of tangential stresses, σ_y		σ_y =					-5250	+2950	+5250	-2950
Shear stress due to load, V_1		$\tau = \frac{V_1}{\pi r_o T}$ =					- 0	0	- 0	+ 0
Shear stress due to load, V_2		$\tau = \frac{V_2}{\pi r_o T}$ =	0	0	0	0				
Shear stress due to Torsion, M_T		$\tau = \frac{M_T}{2\pi r_o^2 T}$ =	0	0	- 0	0	0	0	+ 0	- 0
Add algebraically for summation of shear stresses, τ		τ =								
COMBINED STRESS INTENSITY,		S								
1)	When σ_x & σ_y have like signs:	$S = \frac{1}{2}\left[\sigma_x + \sigma_y + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}\right]$								
2)	When τ = 0	S = largest of σ_x, σ_y, or $\lvert\sigma_x - \sigma_y\rvert$								
3)	When σ_x & σ_y have unlike signs:	$S = \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2}$					-17200			

DESIGN CALCULATIONS	COMPUTED BY LKR	REFERENCE DWGS.
	DATE	
UNION CARBIDE CORPORATION LINDE DIVISION TONAWANDA LABORATORIES TONAWANDA, N Y 14152	CHK'D	SHEET NO 36 OF 46
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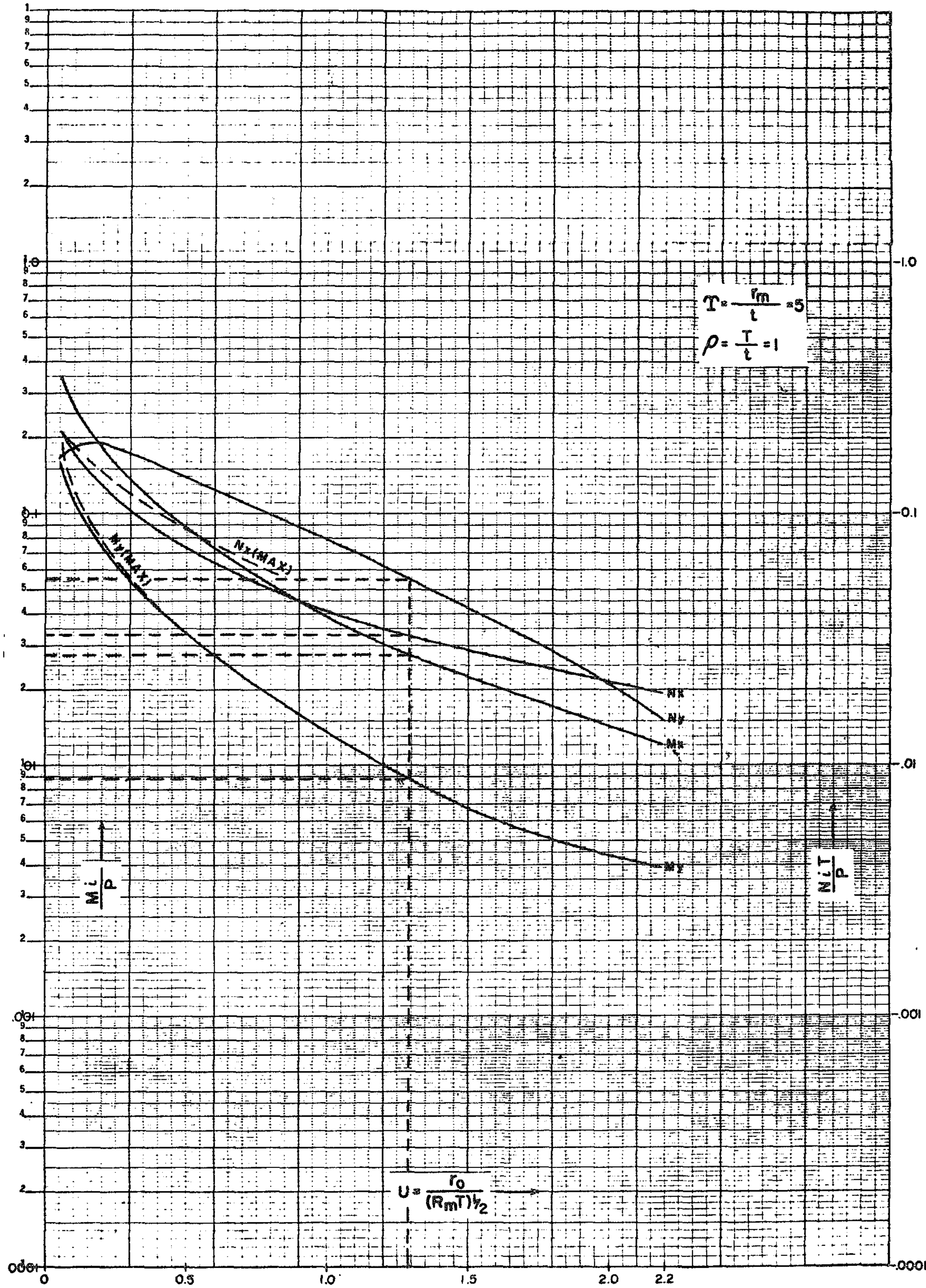

Fig. SP-2—Stresses in spherical shell due to radial load P on a nozzle connection

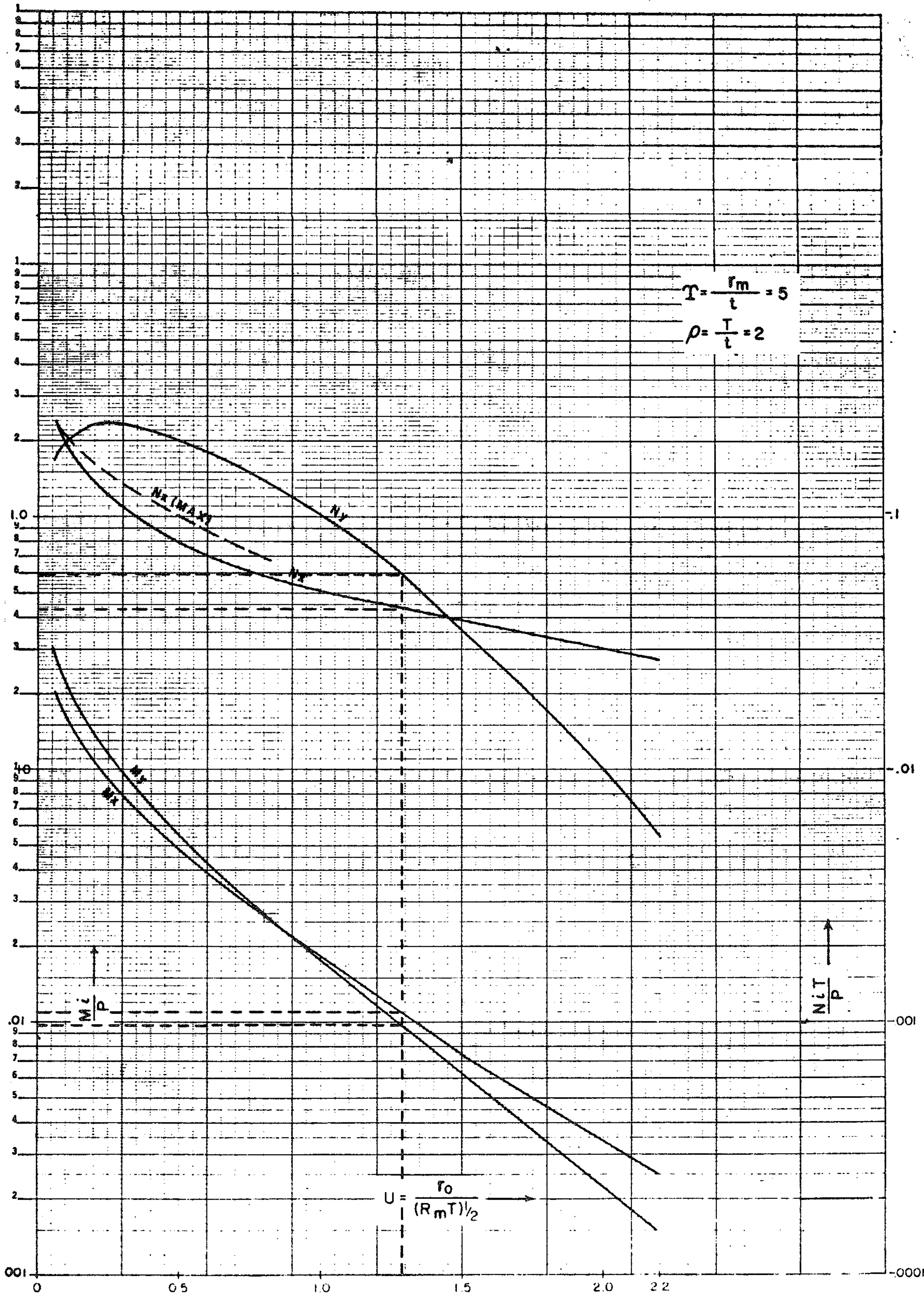

Fig. SP-3—Stresses in spherical shell due to radial load P on a nozzle connection

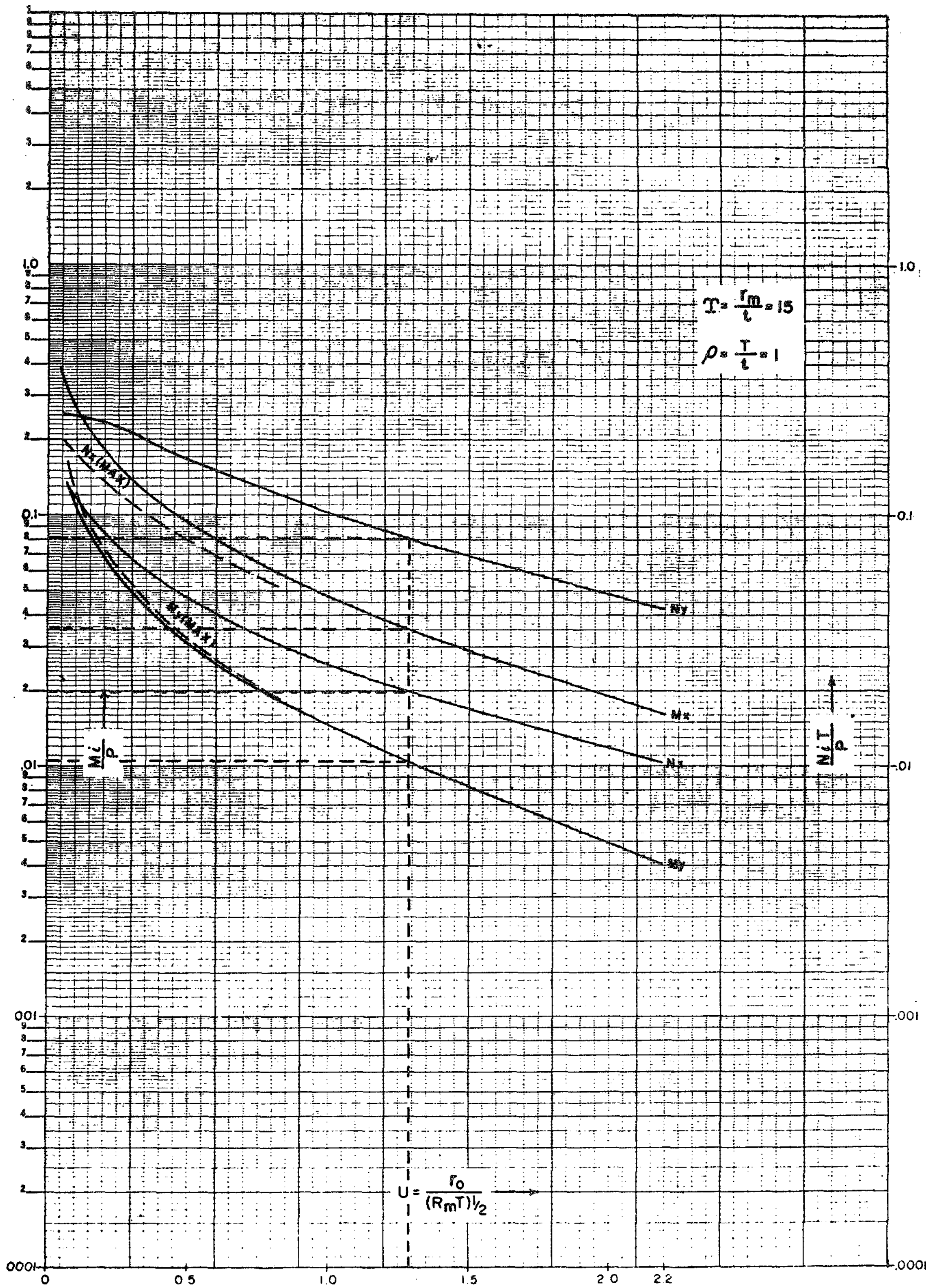

Fig. SP-5—Stresses in spherical shell due to radial load P on a nozzle connection

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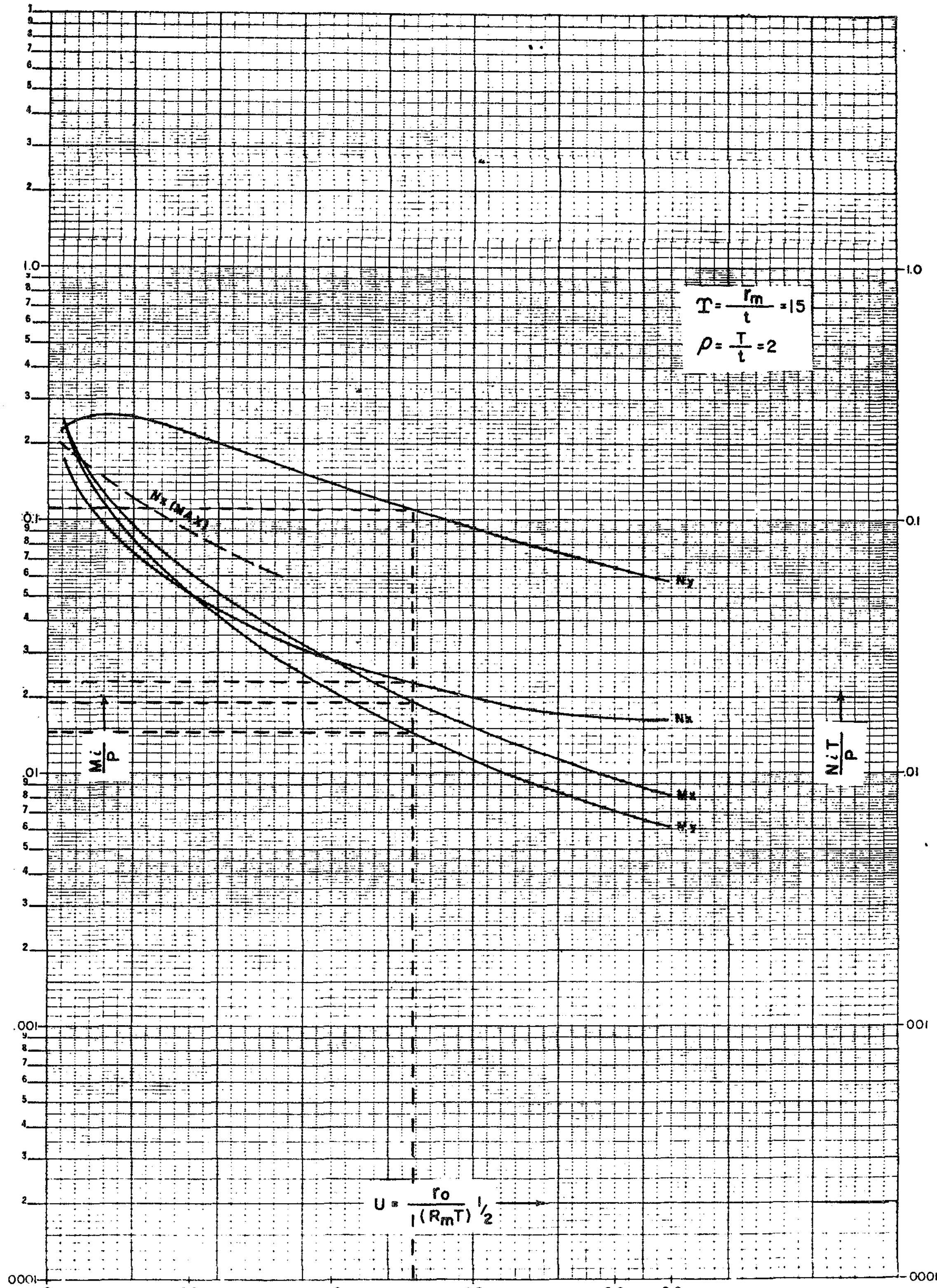

Fig. SP-6—Stresses in spherical shell due to radial load P on a nozzle connection

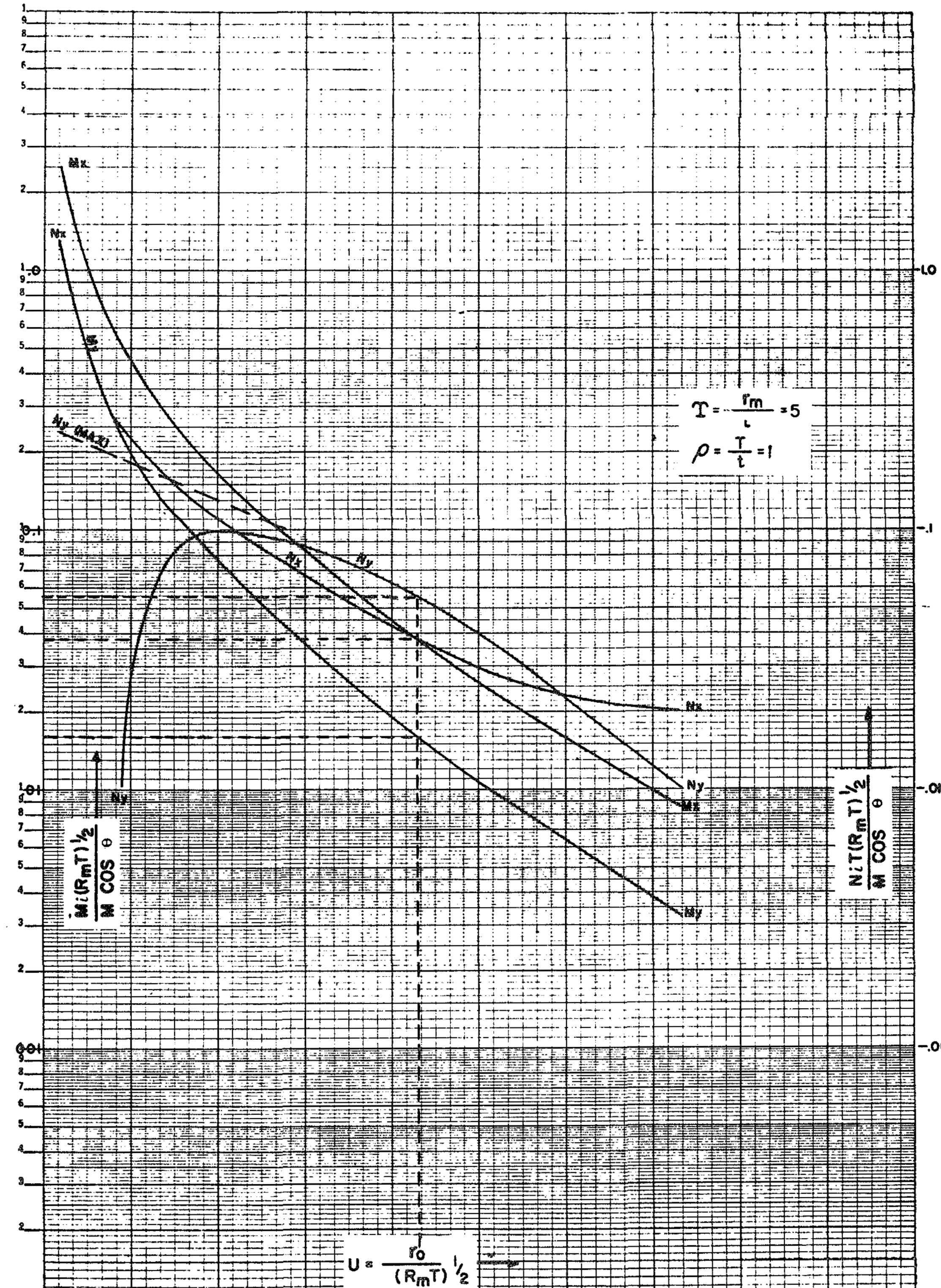

Fig. SM-2—Stresses in spherical shell due to overturning moment M on nozzle connection

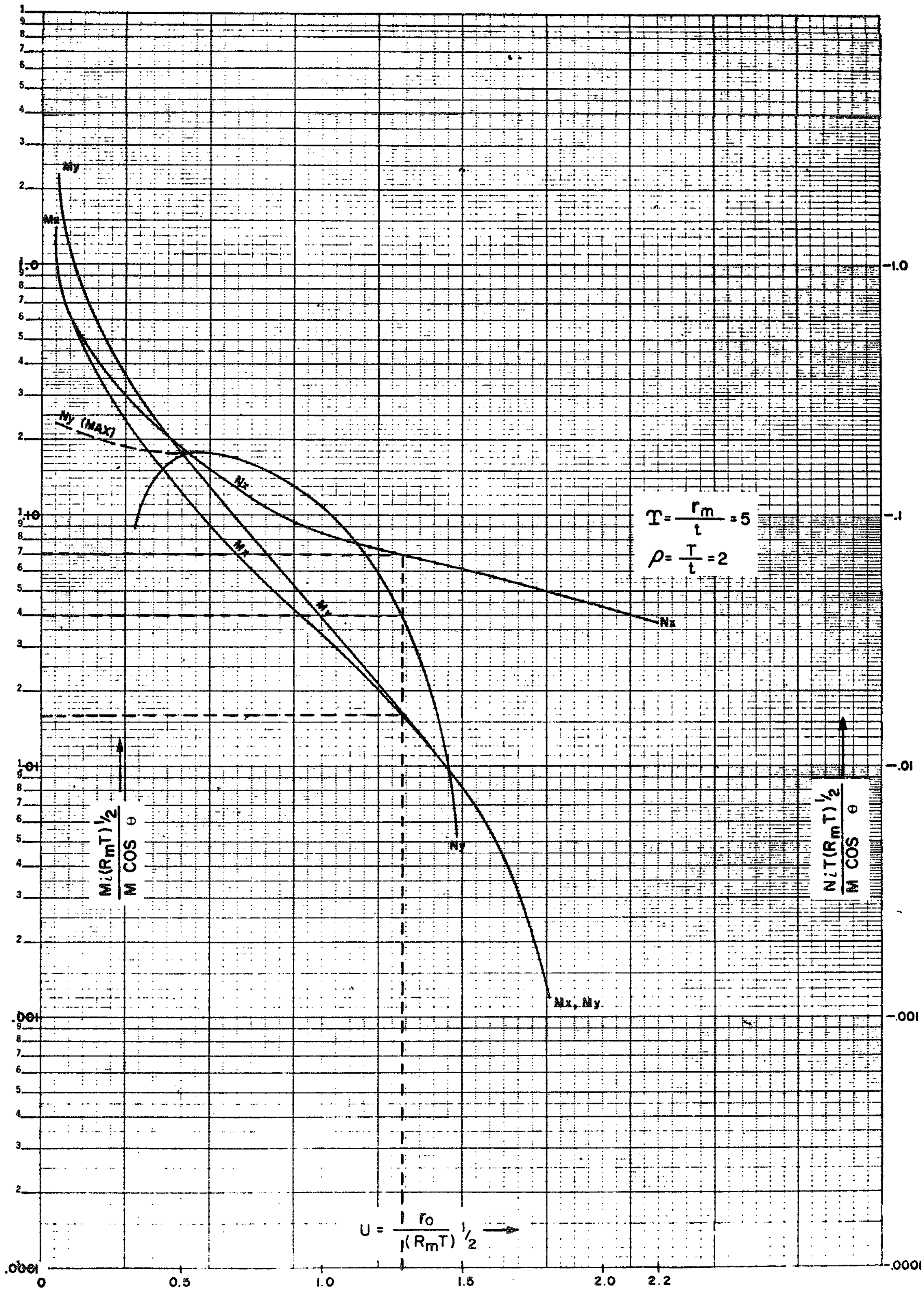

Fig. SM-3—Stresses in spherical shell due to overturning moment M on nozzle connection

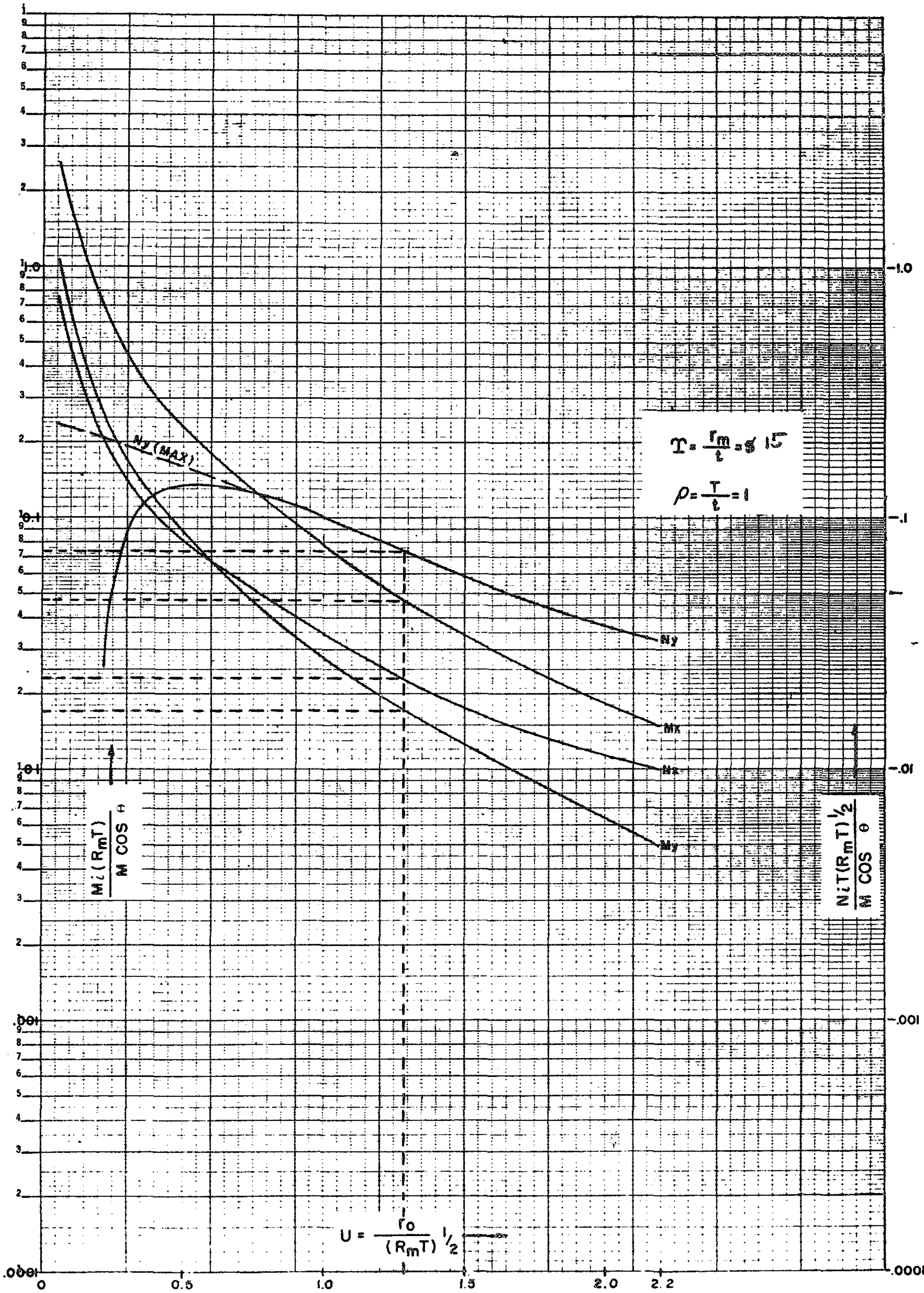

Fig. SM 5—Stresses in spherical shell due to overturning moment M on nozzle connection

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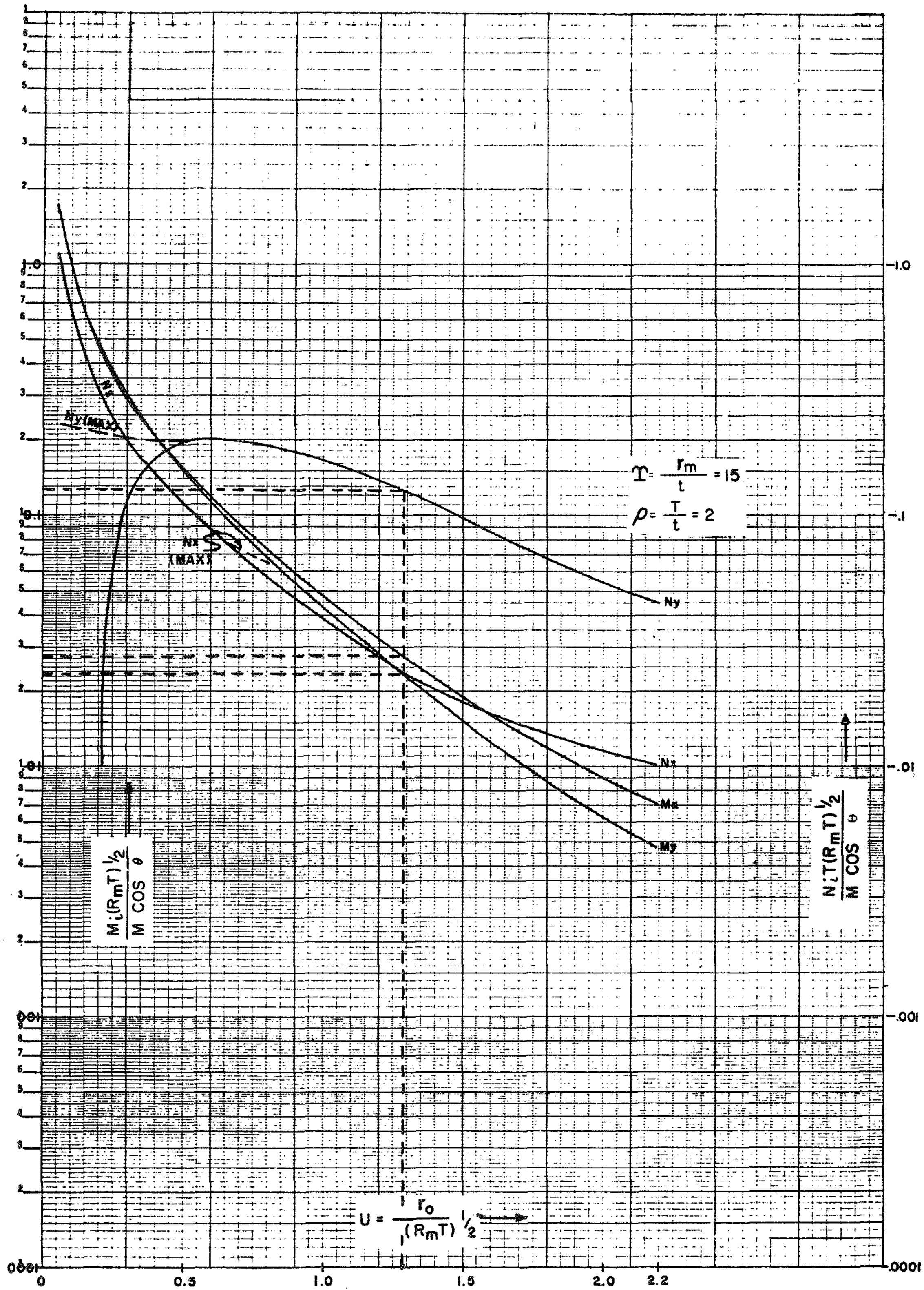

Fig. SM-6—Stresses in spherical shell due to overturning moment M on nozzle connection

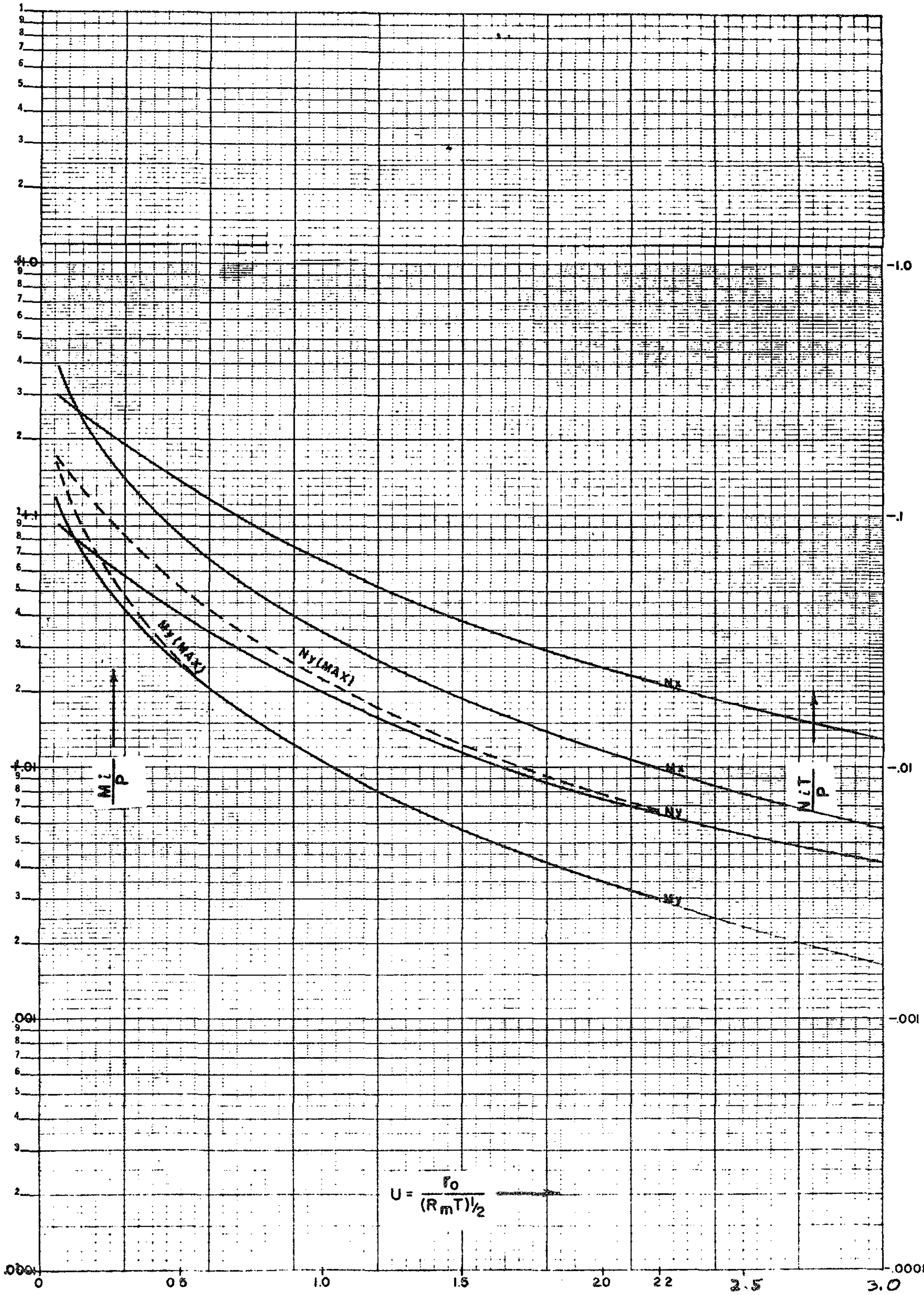

Fig. SR-2—Stresses in spherical shell due to a radial load P on a nozzle connection (rigid plug)

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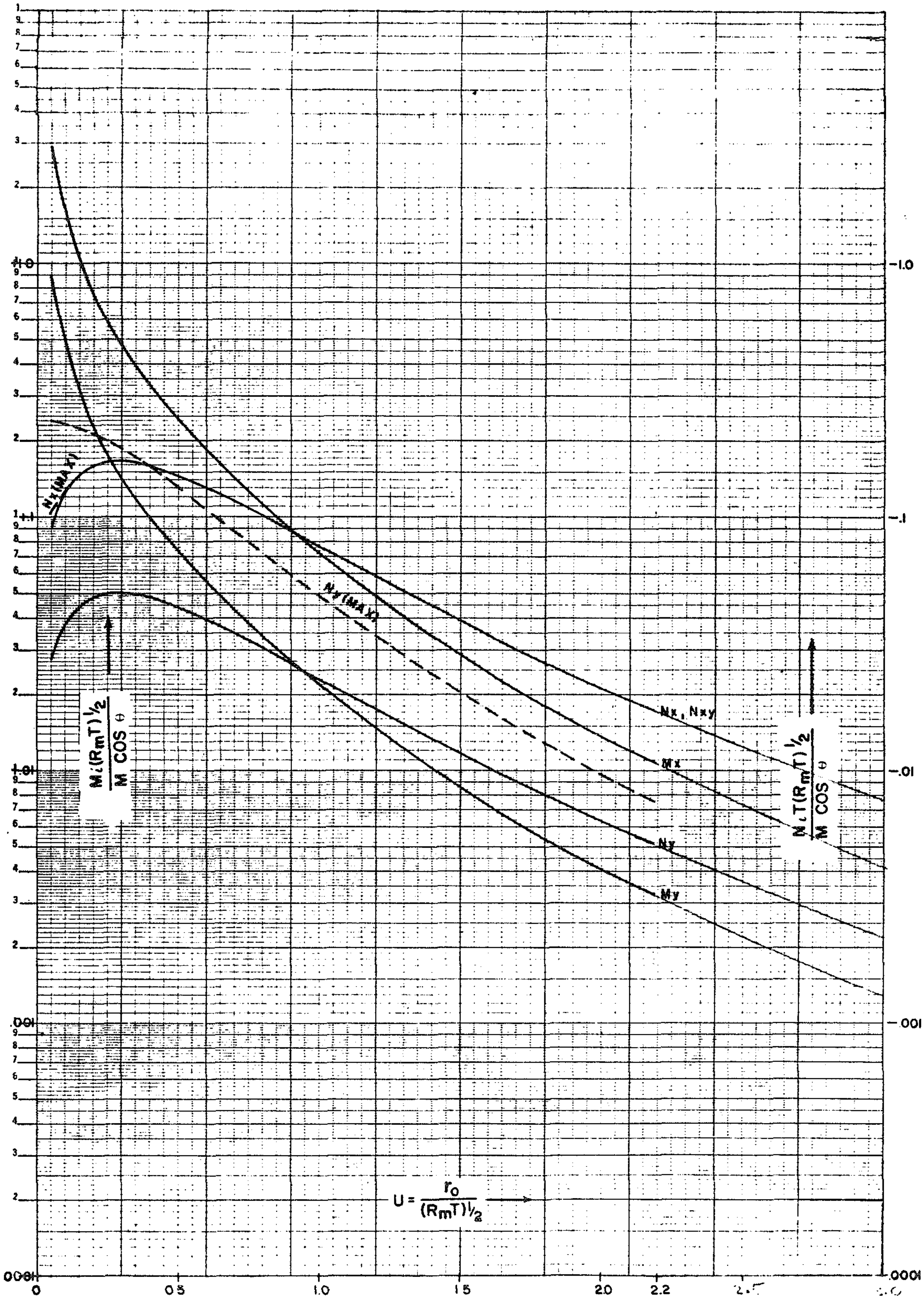
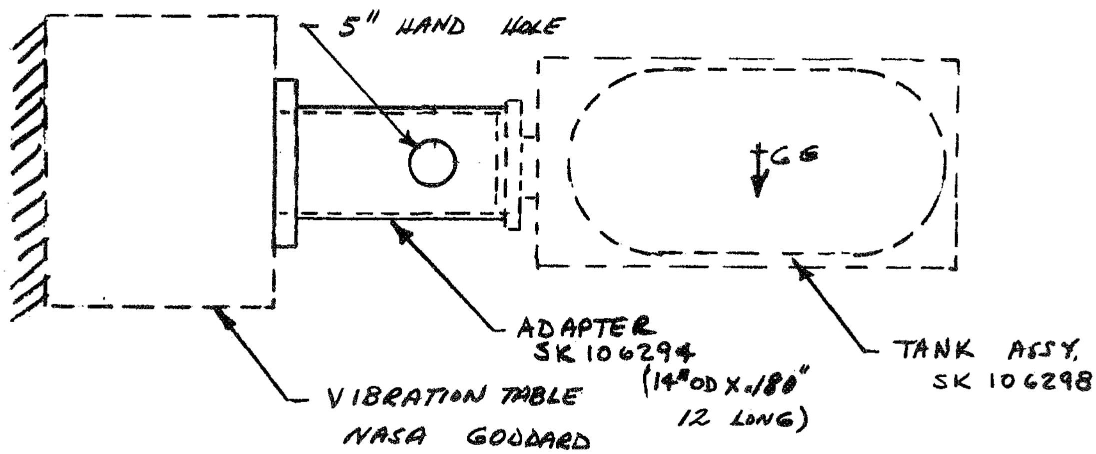


Fig. SR-3—Stresses in spherical shell due to overturning moment M on nozzle connection (rigid plug)

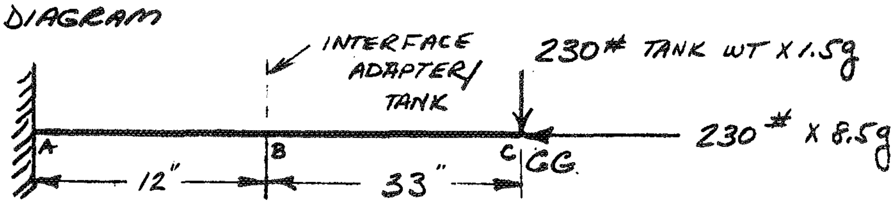
APPENDIX 4

IN ORDER TO LOCATE THE MODEL TANK (SK 106298) ENTIRELY WITHIN THE ACCOUSTIC LINER OF THE LAUNCH PHASE SIMULATOR AT NASA, GODDARD, A SHORT ADAPTER SECTION IS REQUIRED. THE ADAPTER, (SK 106294 SEE FIG. 4) MATES THE TANK TO VIBRATION TABLE VIA THE 18" BOLT CIRCLE (REF. GODDARD DWG GD 1190959). A 5" DIA. HAND HOLE IS PROVIDED FOR ASSEMBLY, AND ENTRANCE WAY FOR THE FILL AND VENT LINES.

INSTALLATION



LOADING DIAGRAM



DESIGN CALCULATIONS CALCULATIONS - ADAPTER VIBRATION TABLE NAS 3-12045	COMPUTED BY GEN	REFERENCE DWGS.: LINDE SK 106294, SK 106298; NASA GD 1090959
	DATE 9/1969 REV. 2/71	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D JM	SHEET NO. 1 OF 8
	APP'V'D	D- APPENDIX 4

FORM 818-323-A

CALCULATION OF LOADS.

POINT (A)

$$\overset{\curvearrowright}{M_A} = (12 + 33)\,230 \times 1.5g = 15{,}500 \text{ IN-\#}$$

$\overrightarrow{P_A}$ = COMPRESSIVE LOAD @ A

$$= 230 \times 8.5g = 1950\ \#$$

$\downarrow V_A$ = VERTICAL SHEAR LOAD @ A

$$= 230 \times 1.5g = 345\ \#$$

POINT B

$$\overset{\curvearrowright}{M_B} = 230 \times 1.5g \times 33 = 11{,}380 \text{ IN-\#}$$

P_B = COMPRESSIVE LOAD @ B

$$P_B = P_A = 1950\ \#$$

V_B = VERTICAL SHEAR LOAD @ B

$$V_B = V_A = 345\ \#$$

MATERIAL

FOR EASE IN HANDLING AND MACHINING, 5083-0 ALUMINUM MATERIAL WILL BE USED.

14" OD X .180 WALL – PERMITS MATING TO TANK PRIOR TO REMOVING TANK FROM TRANSPORTER.

12" LENGTH – REQUIRED LENGTH FOR ACCOUSTIC LOCATION.

DESIGN CALCULATIONS CALCULATIONS ADAPTER – VIBRATION TABLE NAS 3– 12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/1969	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D REV 2/71	SHEET NO. 2 OF 8
	APP'V'D	D- APPENDIX II

FORM 818-323-A

STRESS CALCULATIONS.

BENDING STRESS AT POINT A (TUBE)

$$S = \frac{MC}{I}$$

WHERE $M = 21,750$ in#, $C = 7''$

I_0 = MOMENT OF INERTIA ABOUT NEUTRAL AXIS, ALLOWING FOR THE HAND HOLE

$$I_T = I_{0_{TUBE}} - I_{0_{HOLE}} - AD^2$$

WHERE AD^2 = TRANFER FORMULA FOR MOMENT OF INERTIA.

$= 0$ FOR HAND HOLE LOCATED AT NEUTRAL AXIS

$$I_{HOLE} = \frac{dt^3}{12} = \frac{5 \times (.18)^3}{12} \approx 0$$

$$\therefore I_{TUBE} \approx I_0 = \pi\left(\frac{R_1^4 - R_2^4}{4}\right)$$

WHERE $R_1 = 14/2 = 7$ in

$R_2 = R_1 - .18 = 6.82$

$$\therefore I_{TUBE} = \frac{3.14}{4}(7^4 - 6.82^4)$$

$$= 188 \text{ in}^4$$

DESIGN CALCULATIONS CALCULATIONS ADAPTER – VIBRATION TABLE NAS 3 – 12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D REV 2/71	SHEET NO. 3 OF 8
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FORM 818-323-A

∴ REPEATING BENDING STRESS EQUALS

$$S_A = \frac{MC}{I}$$

$$= \frac{21,750\ \# \times 7}{188}$$

$$= 810\ \text{psi}$$

COMPRESSIVE STRESS AT "A" (TUBE)

CHECK RATIO OF LENGTH / RADIUS OF GYRATION (L/r)

WHERE $L = 12''$

$$r = \sqrt{I/A}$$

OR $I = 188$

$$A = \text{AREA} = \frac{\pi}{4}(D_1^2 - D_2^2)$$

$$= .785\,(14^2 - 13.64^2)$$

$$= 7.85\ \text{in}^2$$

$$\therefore r = \sqrt{I/A} = \sqrt{\frac{188}{7.85}} = 4.9$$

AND $L/r = \frac{12}{4.9} = 2.45$

WHICH INDICATES THAT ADAPTER IS A SHORT COLUMN $(L/r < 50)$ AND ∴ COLUMN FORMULAS ARE NOT REQUIRED

DESIGN CALCULATIONS CALCULATIONS ADAPTER - VIBRATION TABLE NAS 3- 12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D REV 2/71	SHEET NO. 4 OF 8
	APP'V'D	D- APPENDIX II

ORM 815-333-A

CONTINUING, THE COMPRESSIVE STRESS AT A

$$S = \frac{P}{A}$$

WHERE $P = P_A = 1950\#$

$A = 7.85 \text{ in}^2$

$$\therefore S = \frac{1950}{7.85} = 248 \text{ psi}$$

VERTICAL SHEAR STRESS AT A (TUBE)

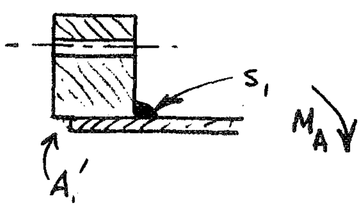
$$S = \frac{2V}{A}$$

WHERE $V = 345\#$

$A = 7.85 \text{ IN}^2$

$$\therefore S = \frac{2(345)}{7.85} = 89 \text{ psi}$$

WELD STRESS AT FLANGE A



SHIGLEY MACHINE DESIGN P. 210

$$S_{ss} = \frac{5.66\, M_A}{\pi\, D^2\, h}$$

WHERE $D = 14$

$h = .18$

$M_A = 21,750 \text{ IN}\#$

$$= \frac{5.66\,(21,750)}{\pi (14)^2 .18}$$

$$= 1110 \text{ PSI}$$

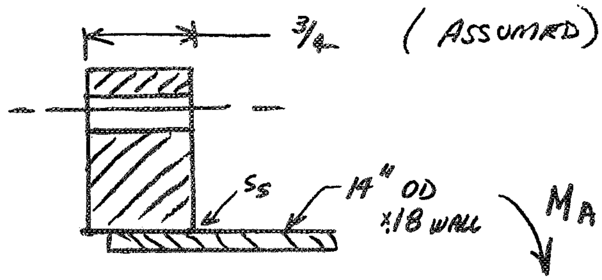
OK

WELD AT A' FOR SAFETY FACTOR

DESIGN CALCULATIONS CALCULATIONS ADAPTER VIBRATION TABLE NAS 3-12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/1969	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 5 OF 8
	APP'V'D	D- APPENDIX II

FORM 818-323-A

RADIAL STRESS IN BOTTOM FLANGE "A"



$M = 15{,}550$ in #.

CASE 5
ROARK p. 195

AT "A" $$S_R = \frac{3 M_A}{4 \pi t^2 r_o}\left[1 + \frac{m+1}{m} \log_e^2 \left(\frac{a - r_o}{K a}\right)\right]$$

WHERE

$r_o = 7$ (FLANGE I.D.)

$a = 10.5$ (FLANGE O.D.)

$\frac{m+1}{m} = 1.3$ M = RECIPROCAL OF POISSONS RATIO

$$K = \frac{.49 a^2}{(r_o + .7a)^2} = \frac{.49 (10.5)^2}{[7 + .7(10.5)]^2} = .262$$

$$S_R = \frac{3 \;\; 15\,550}{4 \pi (.75)^2 \, 7}\left[1 + 1.3 \log_e^2\left[\frac{(3.5)}{.262(10.5)}\right]\right]$$

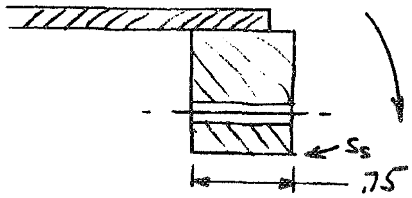
$$= 940\left[1 + 1.3(.932)\right] = 2080 \text{ psi}$$

O.K.

DESIGN CALCULATIONS CALCULATIONS ADAPTER - VIBRATION TABLE NAS 3-12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D REV 2/71	SHEET NO. 6 OF 8
	APP'V'D	D- APPENDIX II

FORM 815-823-A

RADIAL STRESS IN TOP FLANGE B



$M_B = 11{,}380$ IN #.

AGAIN USING CASE 5 ROARK P. 195

$$S_R = \frac{3 M_B}{4\pi t^2 r_o}\left[1 + \frac{M+1}{M}\log_e 2\left(\frac{a-r_o}{Ka}\right)\right]$$

WHERE $a = 7.0$ FLANGE OD.

$r_o = 5.0$ FLANGE ID.

$t = .75$

$M = \frac{1}{\nu} = \frac{1}{.3} = 3.3$

$\therefore \frac{M+1}{M} = 1.3$

$K = \frac{.49a^2}{(r_o + .7a)^2} = .247$

$$\therefore S_r = \frac{3(11{,}380)}{4\pi(.75)^2(5.)}\left[1 + 1.3\left[\log_e \frac{2(7-5)}{(.247)(7)}\right]\right]$$

$= 2000$ psi.

OK.

DESIGN CALCULATIONS CALCULATIONS ADAPTER - VIBRATION TABLE NAS 3-12045	COMPUTED BY GEN	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 7 OF 8
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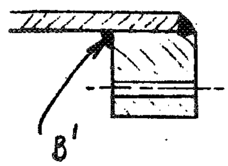
ORM 818-323-A

WELD STRESS AT FLANGE B



$$S = \frac{5.66\,M}{h D^2 \pi}$$ (SHIGLEY p. 210)

WHERE $M = 11{,}380$ in#

$D = 14$

h = WELD FILLET = .18

$$= \frac{5.66\,(11{,}380)}{.18\,(14^2)\,3.14}$$

$= 583$ psi

OK.

∴ WELD AT B' FOR SAFETY FACTOR.

SUMMARY

MAXIMUM STRESSES ARE LESS THAN 3000 PSI. YIELD STRENGTH OF 5083-0 ALUMINUM AS WELDED IS 21,000 PSI THERE FORE DESIGN IS SAFE.

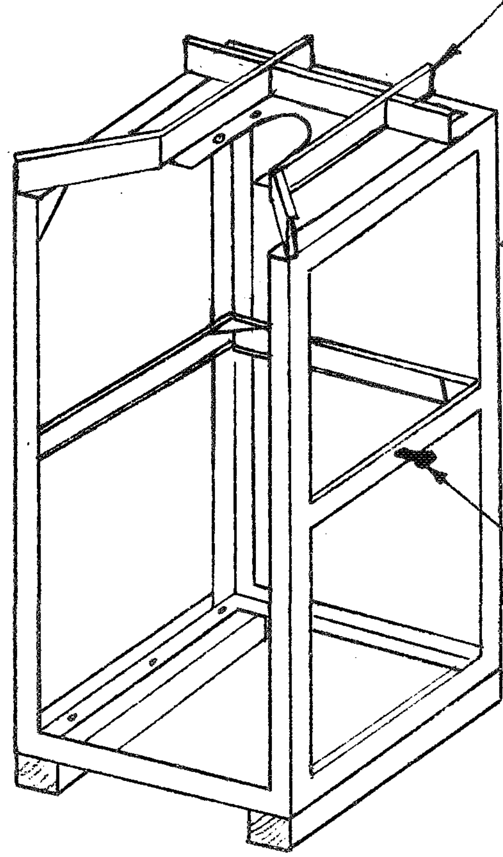
DESIGN CALCULATIONS CALCULATIONS ADAPTER - VIBRATION TABLE NAS3-12045	COMPUTED BY GEN DATE 9/69	REFERENCE DWGS.:
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D APP'V'D	SHEET NO. 8 OF 8 D- APPENDIX II

ORM 818-323-A

APPENDIX 5

THE MODEL TANK MUST BE SUSPENDED (CANTILEVERED) FROM ONE END, WITHOUT SIDE SUPPORTS IN VARIOUS POSITIONS, AND THEREFORE REQUIRES A SPECIAL HANDLING FIXTURE. THE TRANSPORTER (SK 106293 SEE FIG. 5) IS ALSO USED FOR INVERTING THE TANK AT NASA GODDARD.

INSTALLATION



DESIGN CALCULATIONS CALCULATIONS – MODEL TANK TRANSPORTER NAS 3-12045	COMPUTED BY SEM	REFERENCE DWGS.: LINDE SK-106293 SK-106298
	DATE 9/1969	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D JM	SHEET NO. 1 OF 13
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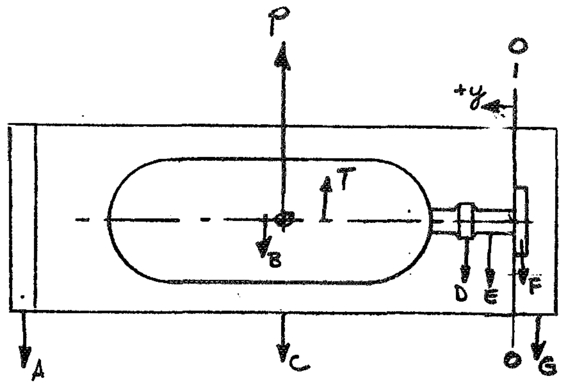
M 815-322-A

LOADING DIAGRAM - TANK MOUNTED IN TRANSPORTER
HORIZONTAL POSITION



DETERMINE CENTER OF GRAVITY AND TOTAL LOADS OF:
(1) CONDITION #1 TANK & TRANSPORTER ASSY.
(2) CONDITION #2. TANK ONLY (ITEMS B, D & E)

LOAD	DESCRIPTION OF ASSY	WEIGHT lbs.	MOMENT ARM AXIS Y-INCH	MOMENT in-lb
A	WOOD BASE	70	80	5600
B	TANK (VESSEL + INSULATION)	129	43	5547
C	BASIC FRAME	367	39	14,313
D	TRANSITION JOINT	45	10	450
E	SUPPORT TUBE	9	3	27
F	SUPPORT FLANGE	50	- 0.5	- 25
G	HEAD STRUCTURE	53	- 2.0	- 106
P	RESULTANT - CONDITION #1	723.0	35.7	25806
T	RESULTANT - CONDITION # 2	183.0	32.9	6024

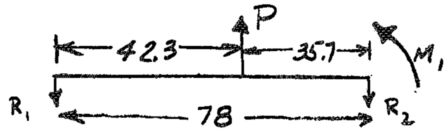
DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS3-12045	COMPUTED BY SSM DATE 9/69	REFERENCE DWGS.:
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D REV 2/71 APP'V'D	SHEET NO. 2 OF 13 D- APPENDIX

F M 615-322-A

CALCULATION OF LOADS CONDITION #1

SIDE RAIL

TANK/TRANSPORTER ASSY. HORIZONTAL



ASSUME BEAM W/ PINNED JOINTS AND END COUPLE M_1

$M_1 = 183^{\#} \times 32.9 = 6020$ IN#

WHERE = $\frac{723}{2 \text{ RAILS}}$

BY SUPER POSITION $P = 361$ #

FOR SIMPLE BEAM ROARK P 102 CASE 12

$$R_1' = \frac{P(42.3)}{78} = \quad = 196 \text{ \#}$$

$$R_2' = \frac{P(35.7)}{78} = \quad = 165 \text{ \#}$$

$$M' = \frac{P(35.7)(42.3)}{78} = 7000 \text{ in\#}$$

FOR END COUPLE ROARK P. 104 CASE 19

$$R_1'' = -\frac{M_1}{78} = -77.3$$

$$R_2'' = +\frac{M_1}{78} = +77.3$$

$$M'' = +M_1 = 6020 \text{ IN\#}$$

$$\therefore R_1 = R_1' + R_1'' = 196 - 77.3 = 118.7$$

$$R_2 = R_2' + R_2'' = 165 + 77.3 = 242.3$$

$$M_{MAX} = M' + M'' = 7000 + 6020$$

$$\therefore M_{MAX} = 13{,}020 \text{ IN\#}$$

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS3-12045	COMPUTED BY DSM	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 3 OF 13
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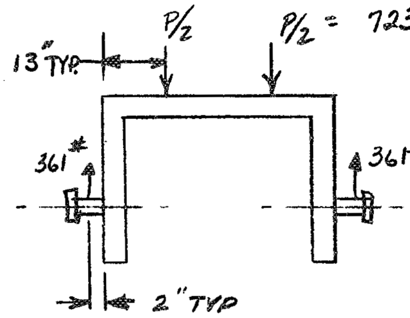
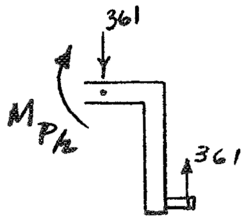
RM 818-328-A

CALCULATION OF LOADS CONTD.
TRUNNION RAIL

ASSY. HORIZONTAL

$P/2 = 723/2 = 361\#$

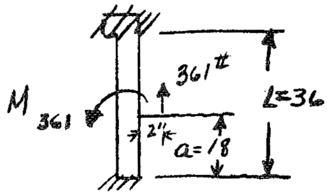




OR $M_{P/2} = 361 \times 15$

$= 5400$ IN #

TRUNNION LOAD = 361 #



ENDS FIXED
TRUNNION LOAD
ROARK P. 109 CASE 37

$$M_{MAX} = M_{361}\left[4\frac{a}{L} - 9\frac{a^2}{L^2} + 6\frac{a^3}{L^3}\right]$$

WHERE $a = 18$

$L = 36$

$$= 2(361)\left[4\frac{18}{36} - 9\frac{18^2}{36^2} + 6\frac{18^3}{36^3}\right]$$

$$= 723\left[2 - 2.25 + .75\right]$$

$$= 723(.5)$$

$$= 361 \text{ IN\#}$$

DESIGN CALCULATIONS CALCULATIONS
MODEL TANK TRANSPORTER
NAS 3-12045

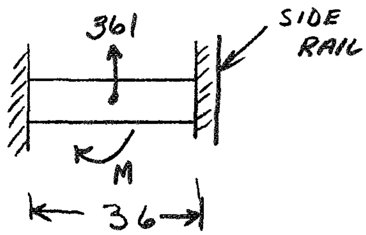
COMPUTED BY SCH
DATE 9/69
CHK'D
APP'V'D
REFERENCE DWGS.:
SHEET NO. 4 OF 13
D- APPENDIX

LINDE COMPANY
DIVISION OF UNION CARBIDE CORPORATION
TONAWANDA LABORATORIES
TONAWANDA, N. Y.

'01 818-323-A

TRUNNION RAIL LOADS (CONTD.)

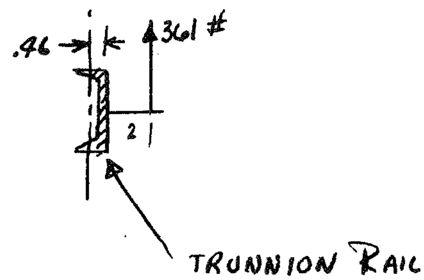
ASSY. VERTICAL



ENDS FIXED
CENTER LOAD
ROARK P. 108
CASE 81

$$M_{MAX} = \frac{361(36)}{8}$$

$$= 1620 \text{ in}\#$$



ENDS FIXED
a) BENDING, LOAD - PIN

ROARK P100 Case 1

$$M = 361(2)$$

$$= 722 \text{ IN}\#$$

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS3-12045	COMPUTED BY SSM	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 5 OF 13
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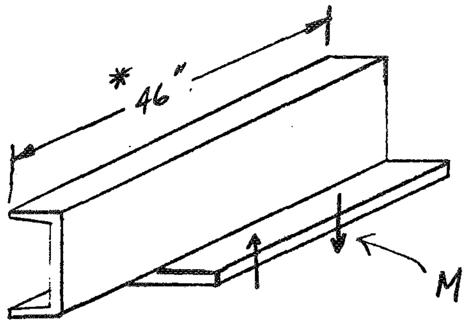
CALCULATION OF LOADS (CONTD.)

HEAD STRUCTURE (SIMPLIFIED)

ASSY VERTICAL WITH 6 G SIDE LOADING



* ASSUMED EQUIVALENT LENGTH FOR HEAD STRUCTURE.

$$M = \frac{T \times Y \times 6g}{2 \text{ RAILS}}$$

$$= \frac{183 \times 32.9 \times 6g}{2}$$

$$= 18,000 \text{ in\#}$$

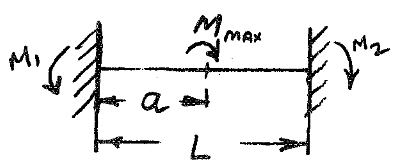
WHERE T = RESULTANT TANK WT CONDITION 2 $= 183\#$

$Y = C.G. = 32.9"$

ASSUME FIXED ENDS WITH CENTER COUPLE

VERTICAL LOADING NOT SIGNIFICANT

ROARK P. 109 CASE 37



$$M_{MAX} = M\left(4\,\frac{23}{46} - 9\,\frac{(23)^2}{(46)^2} + 6\,\frac{23^3}{46^3}\right)$$

$$= 18,000\,(2.0 - 2.25 + .75)$$

$$= 9000 \text{ in\#}$$

WHERE $a = 23$

$L = 46$

$$M_2 = -M_1 = \frac{M}{L^2}\left[2La - 3a^2\right]$$

$$\frac{M}{46^2}\left[2(46)(23) - 3(23)^2\right]$$

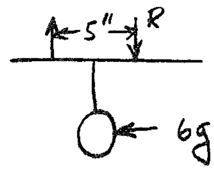
$$= \frac{18000}{2120}\left[2120 - 1590\right]$$

$$= 4500 \text{ in\#}$$

HOLD DOWN BOLTS

FROM HEAD STRUCTURE

$M = 18{,}000$ IN# FOR 2 BOLTS



SIMPLE BEAM

∴ R = LOAD IN ONE BOLT

$$R = \frac{M}{5''}$$

$$= \frac{18{,}000}{5} = 3600\ \#$$

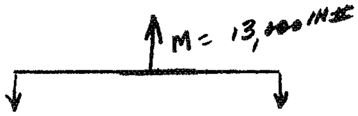
LOAD SUMMARY - CRITICAL CONDITIONS.

	ITEM/ASSY POSITION	LOADS BENDING	
a)	RAIL/HORIZONTAL	13,000 IN#	(PINNED ENDS)
b)	TRUNNION/HORIZONTAL	5400	(FIXED ENDS)
c.)	TRUNNION / VERTICAL	1620	(FIXED ENDS)
d)	TRUNNION PIN/VERTICAL	720	(FIXED END)
e)	HEAD STRUCTURE	18,000	(PINNED ENDS)
		4500	(FIXED ENDS)
f)	HOLD DOWN BOLTS	3600 #	PER BOLT — TENSION.

DESIGN OF MEMBERS

FOR FABRICATION EASE AND MATERIAL AVAILABILITY, STRUCTURAL STEEL SHAPES WILL BE USED, WITH ALL WELDED CONSTRUCTION. AN ALLOWABLE STRESS OF 20,000 PSI.

DETERMINE SIDE RAIL SIZE - SIMPLE BEAM ASSY HORIZONTAL



$$S = \frac{Mc}{I}$$

$$\text{or } I/c = \text{SECTION MODULUS}$$
$$= \frac{M}{S}$$

WHERE $M_{CRITICAL}$ WAS DETERMINED TO BE 13,000 in #

$$\text{OR } I/c = \frac{13000}{20000} = .65 \text{ in}^3$$

FROM STRUCTURAL SHAPE DATA FOR 3 X 3 X 5/16 ANGLE

$\rho = 6.1$ #/ft

$I/c = .71$ in^3

$A = 1.78$ in^2

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS 3-12045	COMPUTED BY: SCM	REFERENCE DWGS.:
	DATE: 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 8 OF 13
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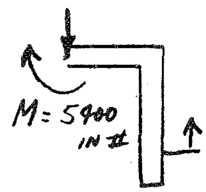
FORM 818-323-A

DETERMINE TRUNNION RAIL
ASSY HORIZONTAL



FOR SIMPLE BENDING

$$S = \frac{Mc}{I}$$

$$\text{or } I/c = \frac{5400}{20,000} = .27 \text{ in}^3$$

FROM STRUCTURAL SHAPE DATA

4 X 1 5/8 X .180 CHANNEL

$I/c = .29 \text{ in}^3$ (MINIMUM)

$A = 1.56 \text{ in}^2$

$\rho = 5.4 \ \#/\text{ft}$

ASSY VERTICAL –

SAME AS ABOVE EXCEPT
LOADING IS IN MAJOR AXIS

FOR 4 X 1 5/8 X .180 CHANNEL

$I/c = 1.9$

$$\therefore S = \frac{1620}{1.9} = 850 \text{ psi}$$

DESIGN CALCULATIONS — CALCULATIONS MODEL TANK TRANSPORTER NAS3-12045	COMPUTED BY: SRW	REFERENCE DWGS.:
	DATE: 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 9 OF 13
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FORM 813-323-A

DETERMINE HOLD DOWN BOLTS

FOR TENSILE FORCE OF 3600 # PER BOLT
AND ALLOWABLE STRESS OF 20,000 PSI

USING $S = \frac{P}{A}$

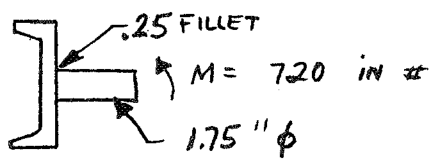
REQD AREA $= \frac{P}{S}$

$$= \frac{3600}{20,000} = .18 \text{ in}^2$$

FROM MECHANICAL DATA
FOR 9/16 - 18 THREAD SYSTEM

THE AREA AT THE MINOR
DIAMETER = .189 IN^2

DETERMINE WELD STRESS AT TRUNNION PIN



FOR BENDING SHIGLEY P 211

$$S = \frac{5.66\ M}{h\ D^2 \pi}$$

WHERE h = FILLET WELD WIDTH
D = PIN DIAMETER

$$= \frac{5.66\ (720)}{(.25)(1.75)^2 \pi} = 1690 \text{ psi}$$

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS 3- 12045	COMPUTED BY SM DATE 9/69	REFERENCE DWGS.:
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D APP'V'D	SHEET NO. 10 OF 13 D- APPENDIX

FORM 815-323-A

DETERMINE TRUNNION PIN

FOR SIMPLE CANTILEVER – BENDING

ASSUMING PIN DIA OF 1.75 INCH

$$S = \frac{Mc}{I} \qquad M = 720 \text{ IN\#}$$

$$= \frac{(720)(.87)}{.46} \qquad I_{o_{1.75}} = \frac{\pi D^4}{64}$$

$$= 1360 \text{ psi} \qquad = \frac{\pi (1.75)^4}{64}$$

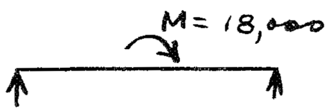
$$= .46$$

WHERE $C = \frac{\text{DIAMETER}}{2}$

$$= \frac{1.75}{2} = .87$$

DETERMINE HEAD STRUCTURE MEMBER

FOR SIMPLE BENDING



$S_{ALL} = 20,000$ psi

$$S = \frac{Mc}{I}$$

$$\text{REQD } I/c = \frac{18,000}{20,000} = .9$$

FOR 4 X 1 5/8 X .160 CHANNEL

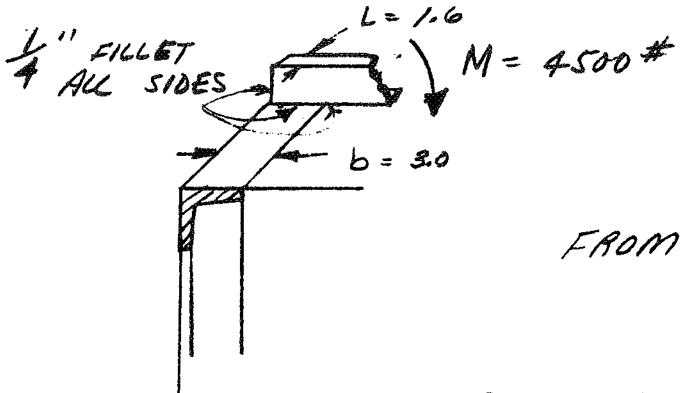
$$I/c = 1.9$$

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS 3-12045	COMPUTED BY SEM	REFERENCE DWGS.:
	DATE 9/69	
	CHK'D	SHEET NO. 11 OF 13
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	APP'V'D	D- APPENDIX

ORM 818-323-A

DETERMINE WELD STRESS AT CORNERS
TOP RAIL TO HEAD

FROM LOAD SUMMARY FOR FIXED ENDS
$M = 4500$ IN #



FROM SHIGLEY
P 211

$$S = \frac{4.24\,M}{h\left[b^2 + 3L(b+h)\right]} \quad \text{STRESS psi}$$

WHERE h = FILLET WELD DEPTH
= .25"

b = SEE FIGURE

L = SEE FIGURE

$$\therefore S = \frac{4.24\,(4500)}{.25\left[3^2 + 4.8[3+.25]\right]}$$
= 3100 psi.

STRESS OK.

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS 3-12045	COMPUTED BY	REFERENCE DWGS.:
	DATE 9/69	
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D	SHEET NO. 12 OF 13
	APP'V'D	D- APPENDIX

TM 818-333-A

SUMMARY

STRESS LEVELS IN ALL MEMBERS ARE LESS THAN 20,000 psi, WHICH WAS THE ASSUMED ALLOWABLE STRESS. YIELD STRENGTH OF STRUCTURAL STEEL IS ~ 45,000 psi.

REFERENCES

1. MACHINE DESIGN J.E. SHIGLEY
McGRAW-HILL BOOK CO.
1956

2. FORMULAS FOR STRESS AND STRAIN
R. J. ROARK McGRAW-HILL BOOK CO.
1954

3. BETHLEHEM STRUCTURAL SHAPES
CATALOG S-58

DESIGN CALCULATIONS CALCULATIONS MODEL TANK TRANSPORTER NAS 3-12045	COMPUTED BY SEU DATE 9/69	REFERENCE DWGS.:
LINDE COMPANY DIVISION OF UNION CARBIDE CORPORATION TONAWANDA LABORATORIES TONAWANDA, N. Y.	CHK'D APP'V'D	SHEET NO. 13 OF 13 D- APPENDIX

ORM 812-323-A

APPENDIX 6

TANK ACCEPTANCE TEST PLAN

CONTRACT NAS 3-12045

A/SK 106462

FOR

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

LEWIS RESEARCH CENTER

CLEVELAND, OHIO

BY

UNION CARBIDE CORPORATION
Linde Division
Tonawanda, New York

G. E. Nies
Project Manager

Revised February, 1970

TANK ACCEPTANCE TEST PLAN

I. OBJECTIVE

In order to verify the validity of the structural analysis of the Model Test Tank, Linde plans to perform the following tests as provided for in the Task II Work Statement.

A. Helium leak test

B. Proof Pressure (hydro) test

C. Static dead load test

1. Axial
   a. Flange down (compression)
   b. Flange up (tension)
2. Cantilever

D. Vibration Test

E. Final Helium Leak Test

The various tests and acceptance requirements are described below. Tests are to be performed in the order listed. The fill and vent lines on the flange assembly (Linde P. N. SK-106415) will be temporarily sealed (by welding) for all tests. Evacuation, proof testing, etc. will be via the two 1/2" NPT Relief valve ports. The relief valves will not be installed during any test. Tank qualification is dependent upon achieving a satisfactory helium leak rate before and after completion of the acceptance test program, unless obvious structural failure occurs. Stress coat (Magna Flux Corporation) and accelerometers will be located on the test tank, per sketch Figures 45, 46, 47 and 48.

II. DESCRIPTION

A. Helium Leak Test

The model tank (Linde P. N. SK-106298) and flange assembly (Linde P. N. SK-106415) will be mated and leak checked as an assembly. The tank will be evacuated with the cryogenic fill and vent lines temporarily sealed. (Prior to final assembly, the dissimilar metal joint/head transition piece weld will be helium leak checked.) An acceptable total leak rate is 1×10^{-7} std cc air per sec or better.

B. Proof Pressure (Hydro Test)

The model tank (Linde P N. SK-106298) and flange assembly (Linde P. N. SK-106415) will be evaluated using clean water and suitable pressurizing equipment to achieve a test pressure of 115 psig. (1-1/2 times the 75 psid internal pressure requirement.)

C. Static Dead Load Tests

1. Axial (Vertical) Static Test

(a) The model tank will be tested in a flange down tank axis vertical position, thereby loading the support pipe in compression. The tank weight will be increased 50% over the anticipated 8.5 g loading (2.5 g acceleration, $\pm$ 6 g vibration) during dynamic testing.

(b) The model tank will be tested in a flange up tank axis vertical position, thereby loading the support pipe in tension. The tank weight will be increased 50% over the anticipated 3.5 g loading (+ 2.5 g acceleration plus the - 6.0 g vibration, or 3.5 g tension during dynamic testing).

2. Cantilever (Horizontal) Static Test

The model tank including vibration adapter, will be tested in a tank-axis-horizontal position, thereby submitting the support pipe to a bending load. Tank weight is to be increased by 50% over that expected during testing of model tank 1/2 full of liquid nitrogen at a 1 g loading.

D. Vibration Test

The model tank, flange assembly, and vibration adapter will be vibration tested in a tank axis horizontal position using a horizontal slide table with end input. The tank will be cantilevered off of the flange end. Test range will be from 20 to 150 cps, to a maximum 10.0 g sinusoidal vibration level in the test axis. Maximum acceleration in any other axis not to exceed 5.0 g.

E. Final Helium Leak Test

A second helium leak test will be performed after all other proof tests, static and dynamic, are completed. An acceptable total leak rate will be 1×10^{-7} std cc air per sec.

III. TEST SET-UP

A. Helium Leak Test

The model tank (Linde P. N. SK-106298) mated to flange assembly (Linde P. N. SK-106415) will be leak checked as an assembly. The assembled unit can be suspended from the model tank transporter (Linde P. N. SK-106293) during test. The unit will be connected to a leak detector, and a helium leak rate will be determined after the system has been evacuated to 100 microns of mercury absolute or less. A bag test whereby the entire evacuated unit is surrounded by helium gas (contained in a plastic bag) will be employed.

B. Proof Pressure (Hydro) Test

The model tank (Linde P. N. SK-106298) and flange assembly (Linde P. N. SK-106415) will be hydrotested as an assembly. The assembled unit will be up-ended in the model tank transporter (Linde P. N. SK-106293) during the test. Fluid taps will be via the relief valve ports (1/2" NPT).

C. Static Dead Load Tests

1. Axial (Vertical) Static Test

(a) The model tank, flange assembly and vibration adapter assembly will be mounted in the model tank transporter (Linde P. N. SK-106293), inverted to a flange down position, and lowered to the floor. Suitable stands of appropriate height should be placed under the transporter head frame work for support to prevent tipping. Primary tank support will be achieved through the adapter section (Linde P. N. SK-106294) resting on the floor.

The test weight on the support tube is to be increased to 1.5 times the empty tank weight when subjected to a 8.5 g loading. (The 8.5 g loading is arrived at by considering the 6.0 g vibration level plus a 2.5 g acceleration level per contract specifications. From "Summary of Weights" Progress Report No. 5, Appendix I, page 9 of 46, the applicable tank weight consists of items E, F and G for a total of 136 pounds.

Therefore, the total required support tube loading is 136 lbs. x 1.5 x (2.5 + 6.0 g) = 1730 lbs.

A summation of the tank weight (136 lbs.) and the weight of the contained water - 1173 lbs. (140 gallons) yields an available total weight of

$$136 + 1173 = 1309 \text{ lbs.}$$

Thus, subtracting the available weight (1309 lbs.) from the required loading (1730 lbs.) yields a required add on weight of

$$1730 \text{ lbs. minus } 1309 \text{ lbs.} = 421 \text{ lbs.}$$

(b) The model tank and flange assembly will be mounted in the model tank transporter in a flange up position. The test weight on the support tube is to be increased to 1.5 times the empty tank weight when subjected to the anticipated 3.5 g loading. (The applicable tank unit is 136 lbs., see section III, C, 1, a.) Therefore, the total required support tube loading is

$$136 \text{ lbs. x } 1.5 \text{ x } (2.5 \text{ g} - 6\text{g}) = 715 \text{ lbs. tension}$$

Thus, subtracting the empty tank weight from the total required loading,

$$715 - 136 = 579 \text{ lbs. water}$$

it is determined that 579 lbs. of water must be added to the tank. Therefore, it will be necessary to add 70 gallons (585 lbs.) of water to the tank.

2. Cantilever (Horizontal) Static Test

The model tank, and vibration adapter, will be tested in a tank-axis-horizontal position.

The test load on the support tube is to be increased to 1.5 times tank weight, when 1/2 full of liquid nitrogen. (Expected dynamic loading in direction normal to tank axis is 1 g due to gravity.)

Therefore, the total required support tube loading is 136 lbs. (tank) plus 70 gallons of liquid nitrogen (140 gallons total capacity) times 1.5 which is 1.5 (136 + 70 (6.7 lbs./gallon)) = 908 lbs.

Thus, subtracting the empty tank weight from the total required loading,

$$908 - 136 = 772 \text{ lbs.}$$

it is determined that 772 lbs. of water must be added to the tank. Therefore, it will be necessary to add 93 gallons (775 pounds) of water to the tank.

D. Vibration Test

The model tank, flange assembly, and vibration adapter will be vibration tested in a tank-axis-horizontal position only. The unit will be attached to a horizontal slide table, which will be driven by the vibration machine. The model test tank unit will be cantilevered from the flange end, thus producing the same vibration loading as that expected during testing on the Launch Phase Simulator at NASA Goddard. The test range input will be extended to 10.0g sinusoidal in the test axis, thus exceeding the expected vibration loading of 6.5 g plus 2 g acceleration. (Acceleration in any other axis not to exceed 5.0 g.) The frequency range will be varied between 20 and 150 cps. (Total assembly weight is ~ 300 lbs., overall package dimensions are ~ 34 in. diameter by ~ 85 in. long.)

Two tri-axial accelerometers will be used. One of the accelerameters will be mounted on the model tank flange, and the second accelerometer will be mounted at the center of gravity. The accelerometer mounted on the model tank flange shall be used to monitor and control the vibration input. Crosstalk between the three axes of the unit will be observed on the accelerometer placed at the tank center of gravity.

E. Final Helium Leak Check

The second and final helium leak rate will be determined after all other proof tests are completed. For description, see details as described in Section IIIA.

IV. TEST PROCEDURE

A. Helium Leak Check

(1) Place the model test tank (Linde P. N. SK-106298), including flange assembly SK-106415) in the model tank transporter (Linde P. N. SK-106293). (Install a new gasket, 0.125" thick 1100 series aluminum, between the model tank and flange assembly and torque bolts to a minimum of 100 ft. lbs.) Connect a Veeco MS-9 helium leak detector to the tank via one of the relief valve ports (1/2" N.P.T.), and plug the second port (also 1/2" N.P.T.) with a pipe plug.

(2) Evacuate the test enclosure below 100 microns pressure.

(3) The leak detector scale is zeroed.

(4) The standard leak unit mounted on the Veeco unit is opened into the system, causing the leak detector scale to indicate the number of units proportional to the standard leak.

(5) Record this value of scale reading for calculation of leak rate.

(6) Valve off the standard leak unit from the system and note that the detector scale returned to zero.

(7) Enclose the area to be leak tested in a plastic envelope containing helium at atmospheric pressure.

(8) Record the steady-state value of the scale reading for calculation of leak rate.

(9) Calculate leak rate.

$$\text{Leak Rate } \frac{\text{std-cm}^3 \text{ air}}{\text{sec}} = \frac{\text{steady state detector scale reading (unit)}}{\text{standard leak detector scale reading (unit)}} \times$$

$$\text{Value of standard leak } \left(\frac{\text{std-cm}^3 \text{ air}}{\text{sec}}\right)$$

Sample of Calculations:

Given:

Steady state detector scale reading = 20 (unit)
Standard leak detector scale reading = 6 (unit)
Value of standard leak = 2.8×10^{-8} std.cc air/sec.

$$\text{Leak rate} = \frac{20 \times 2.8 \times 10^{-8}}{6} = 9.3 \times 10^{-8} \frac{\text{std-cm}^3 \text{ air}}{\text{sec}}$$

B. Proof Pressure (Hydro) Test

(1) Install the model test tank P N. SK-106298, including flange assembly SK-106415 and vibration adapter assembly SK-106294 in the model tank transporter SK-106293. See Figure 45.

(2) Install a vent line through one of the two 1/2" NPT ports in the flange assembly. Cut the inserted end of the vent line at an angle to assure an open passage when the vent line is bottomed against the tank. Install a fill and drain through the second 1/2" NPT port on the flange assembly. Extend both the vent and the fill/drain lines through one of the 5" diameter access holes on the vibration adapter.

(3) Invert the transporter/tank assembly, into a flange down position, tank axis vertical, and lower to the floor. Primary support is to be obtained through the vibration adapter. Place stands of appropriate height at the corners of the transporter to achieve stability. Tank must be level.

(4) Apply stress cost per sketch, Figure 45.

(5) Fill the tank with clean tap water, noting quantity by use of a totalizing water meter. Fill tank until water is observed flowing from the vent valve.

(6) Close the vent valve.

(7) Attach pressurizing equipment and slowly raise pressure to 115 psig (minimum pressurizing time - 2 minutes). Maintain pressure for 15 minutes, observe and record stress coat analysis.

(8) Release pressure and drain tank.

C. Static Dead Load Tests

1. Axial (vertical) Static Test

(a) Tank Assembly - flange down

1. Tank assembly remains in the inverted position after completion of the proof pressure test. That is, the tank assembly is mounted in the transporter, inverted and setting on the floor, with primary tank support being achieved via the vibration adapter. Tank plumbing is also the same as that used in the proof pressure tests.

2. Apply stress coat per sketch (Figure 45).

3. Construct a temporary wood platform within the head/skirt structure, fill tank with water, and add necessary weights. The weight of the platform and add on weights is to total a minimum of 421 pounds.

4. Observe and record stress coat analysis.

5. Remove extra weight and platform, and drain the water from the tank.

(b) Tank Assembly - flange up

1. The model test tank and flange assembly is mounted in the tank transporter flange up. Tank support is achieved entirely via the flange bolted to the transporter. Tank plumbing is the same as that used in the proof pressure test, except that the fill and vent functions are reversed.

2. Apply stress coat per sketch Figure 46.

3. Add 70 gallons of water to the test tank as measured by a totalizing water meter.

4. Observe and record stress coat analysis.

5. Drain the water from the tank by pressurizing the vent line, and withdrawing via the fill line. Do not exceed 10 psig tank pressurization.

2. Cantilever (axis horizontal) Static Test

(a) Position model Tank/flange assembly/vibration adapter to a horizontal position and attach flange to support. (The flange contains 10 holes - 7/8" diameter on 18" bolt circle.

(b) Reposition the temporary vent line inserted through the 1/2" NPT port to contact the top of the cylindrical portion of the tank. This will allow filling the tank in the horizontal (test) position. Add a fill line through the remaining 1/2" NPT port that contacts the bottom of the cylindrical portion of the tank in test position. (This line will be used to pressure drain the tank at the completion of the test.)

(c) Apply stress coat per sketch (Figure 47).

(d) Add 93 gallons of water (775 pounds) to the test tank as measured by a totalizing water meter.

(e) Observe and record stress coat analysis.

(f) Drain the tank by pressurizing the vent line, and withdrawing via the fill line. DO NOT EXCEED 10 PSIG TANK PRESSURIZATION.

(g) After the tank has been emptied, remove the temporary fill and vent lines, install the tank and vibration adapter in the transporter, and invert the tank/transporter to a flange down position.

(h) Insert a 1/4" diameter line from an air heater through one of the ports, and purge overnight with 250°F air.

D. Vibration Test

(1) Place assembly of vibration adapter SK-106294, flange assembly SK-106415 and model test tank SK-106298 onto vibration machine as shown in Figure 48.

(2) Testing will be at ambient pressure and temperature.

(3) Locate two triaxial accelerometers as indicated on Figure 48. (Input control via flange accelerometer.)

(4) Perform the required vibration test as per the following, (Total test time 30 minutes) - record accelerometer data.

(a) 20 → 150 → 20 Hz cycled logarithmically from minimum to maximum in 7-1/2 minutes at 1 g level. Note resonance points but do not dwell.

(b) Repeat above at 10.0 g level. (Maximum 11.0 g.) (NOTE: ACCELERATION IN ANY OTHER AXIS NOT TO EXCEED 5.0 g.)

E. Final Helium Leak Test

(1) Repeat helium leak test as per Procedure of Section IV, item A.

V. TEST EQUIPMENT

A. Helium Leak Test

1. Helium Leak Detector required sensitivity of 1×10^{-11} standard cc per second air.

2. Helium/air standard leak.

B. Proof Pressure (hydro) Test

1. Hydro Test equipment similar to

Hydro Tester Model 9118 A
Watson Stillman Pressure Division
Roselle, New Jersey

2. Calibrated test gage - range 0 → 150 psig.

3. Stress coat - Magnaflux Corporation, Kit 102A, Chicago, Illinois.

4. Totalizing water meter similar to

Buffalo Meter Co., C size, 100,000 gallon, 0 - 10 gallon scale - 1 gallon increment.

C. Static Dead Load Tests

1. Stress coat.

2. Totalizing water meter (same as above).

D. Vibration Test

1. Vibration Testing Equipment capable of performing a vibration test as specified in Section IIID. Similar to M. B. Electronics, Model C210.

2. Accelerometers and readout equipment similar to Triaxial Accelerometers composed of three MB Electronics Model 302 and related readouts.

E. Final Helium Leak Test

Repeat Test - same instrumentation as used and described in Section V-A.

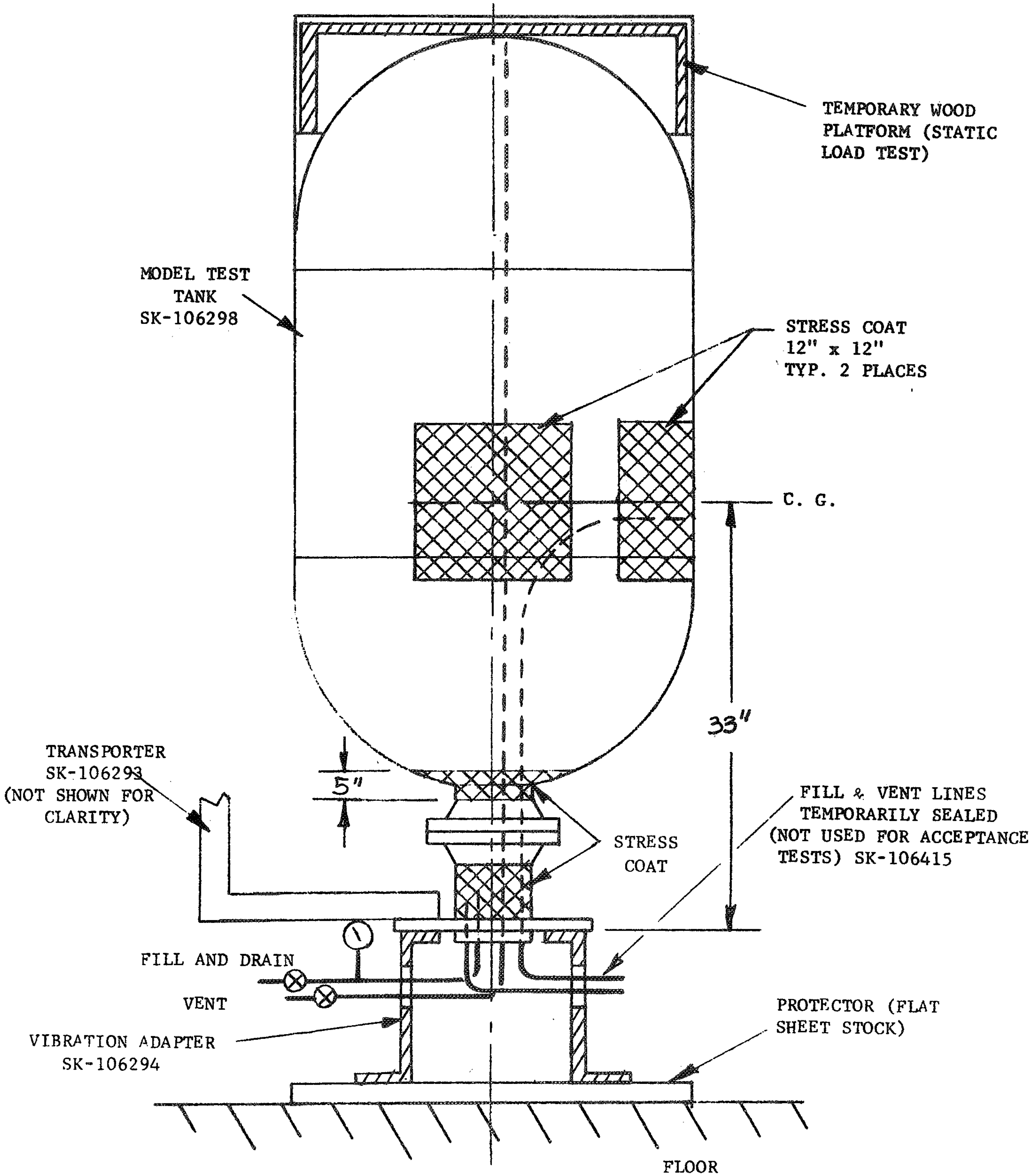


FIGURE 45 TEST SET UP - PROOF PRESSURE / AXIAL LOAD

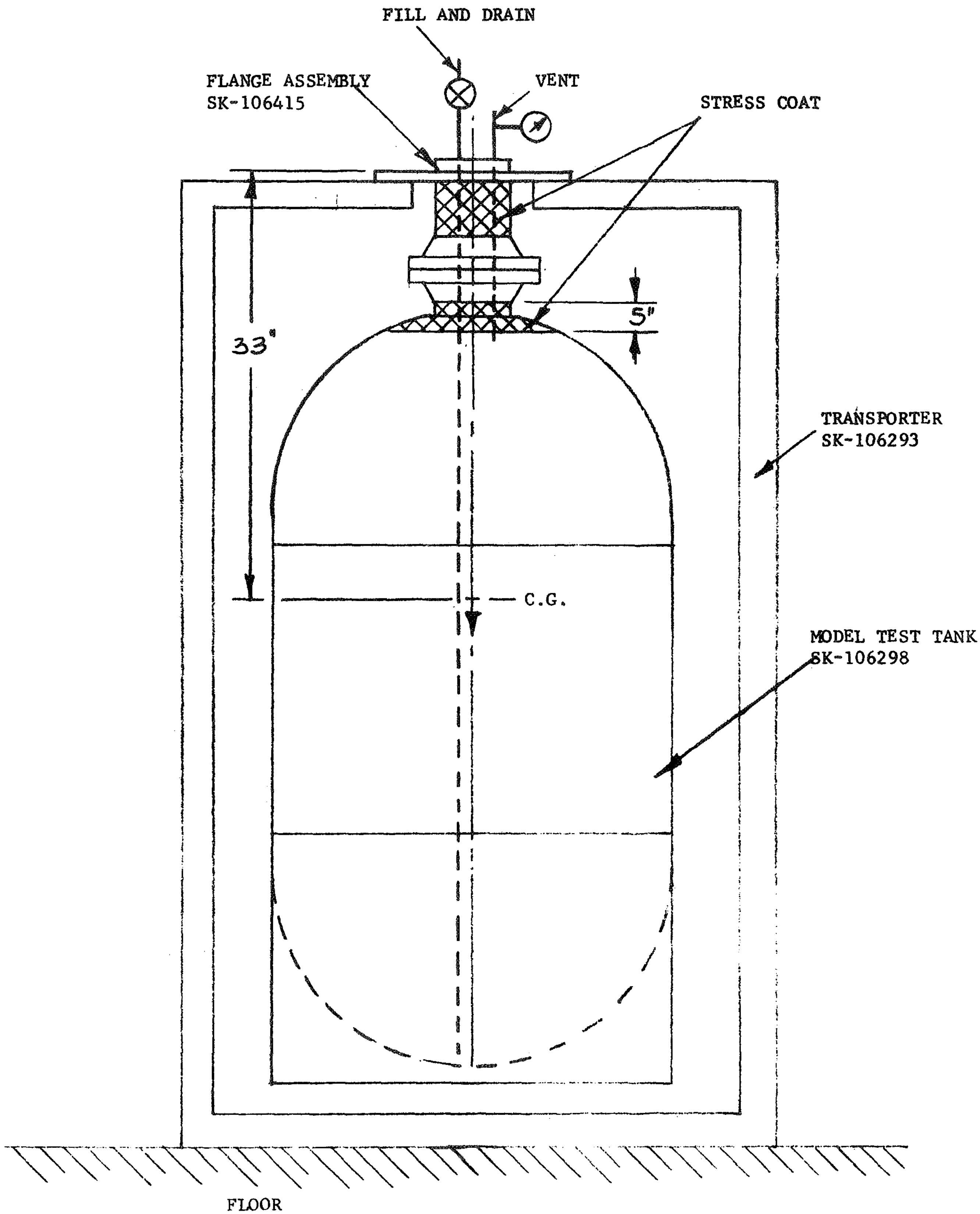


FIGURE 46 TEST SET-UP (AXIAL LOAD) FLANGE UP

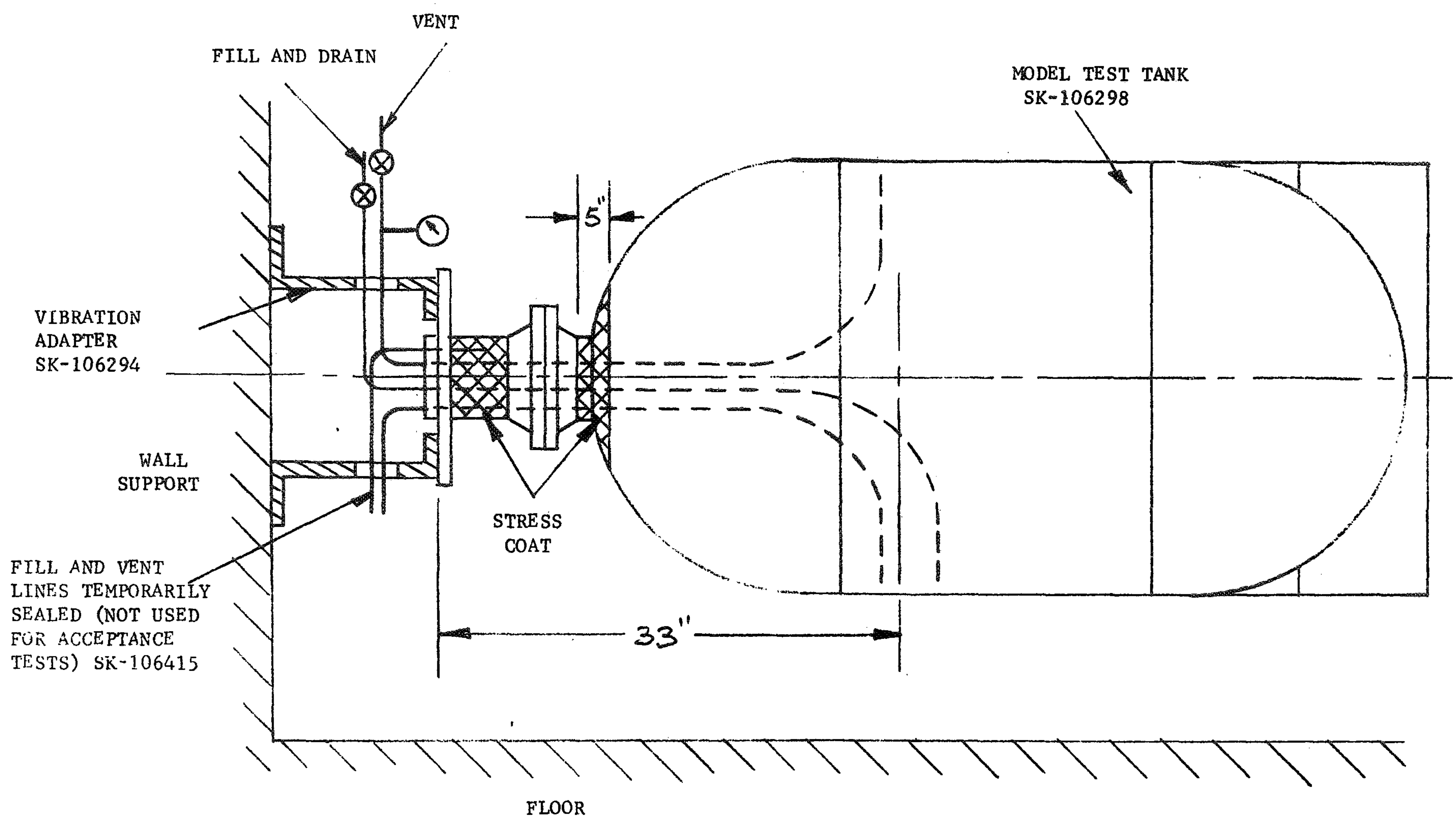


FIGURE 47 TEST SET-UP - CANTILEVER (AXIS HORIZONTAL)
DEAD LOAD TEST

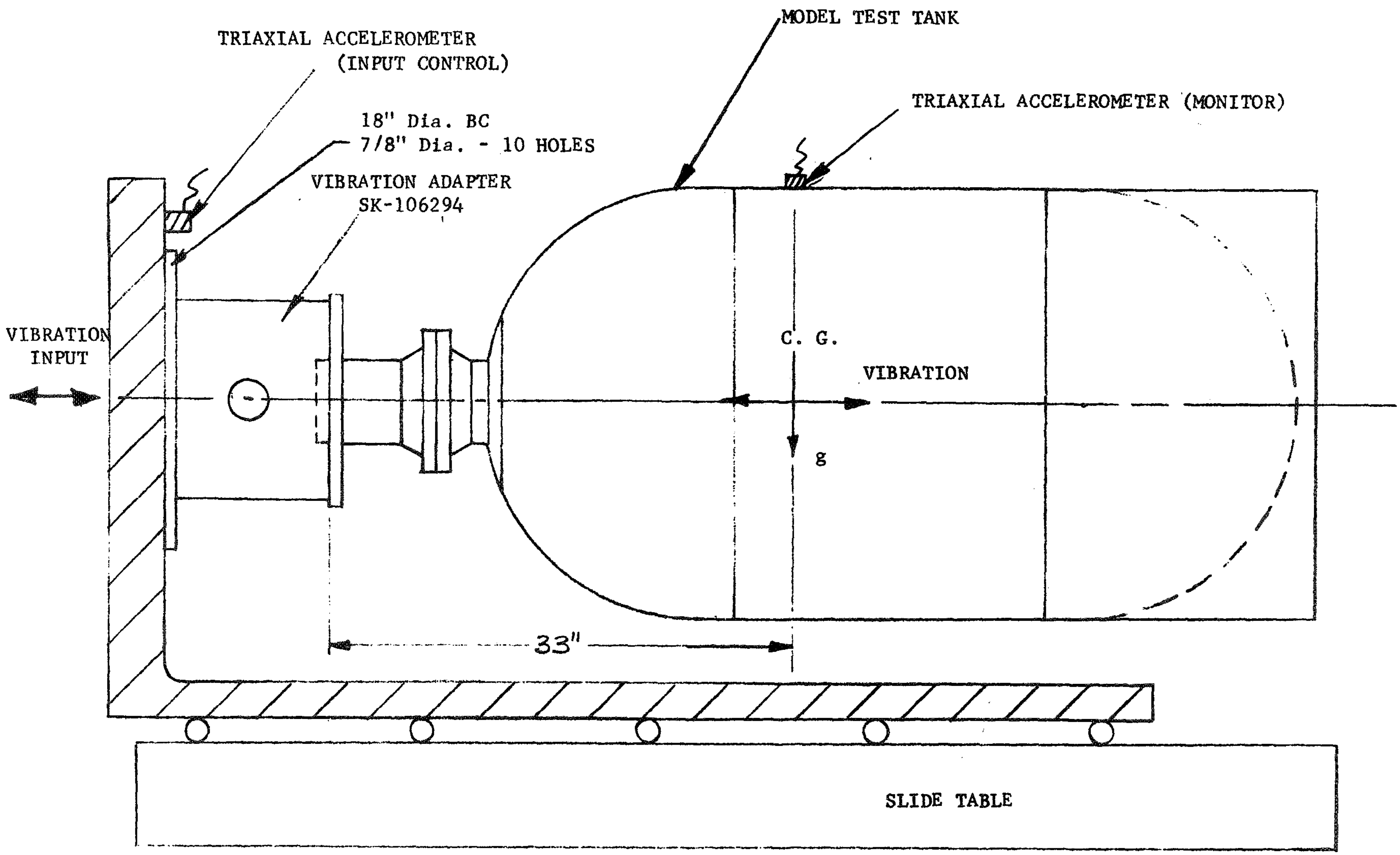


FIGURE 48 TEST SET UP - VIBRATION TEST

APPENDIX 7

DAYTON T. BROWN INC.

CHURCH STREET • BOHEMIA, LONG ISLAND, NEW YORK 11716

AREA CODE 516 589-6300

22 April 1970

DTB03R70-0587
J/N 8565

Union Carbide Corporation
Linde Division
P.O. Box 44
Tonawanda, New York 14150

Attention: Mr. J. Neville

Subject: Dynamic Testing of One (1) Tank, S/N 629

References:
(a) Union Carbide Corp. Purchase Order Number 825L43514
(b) Dayton T. Brown, Inc. Job Number 8565
(c) Union Carbide Corp. Test Plan, dated February 1970

Enclosures:
(1) Test Equipment List
(2) Vibration Test Graphs
(3) Typical Setup Photograph

Gentlemen:

This report presents the results of a Vibration Test performed on the above subject as requested in reference (a).

The test item arrived at Dayton T. Brown, Inc. on 14 April 1970. Testing was started on 15 April 1970 and was completed on 15 April 1970. The test item was returned to Union Carbide Corp. on 15 April 1970.

Present during the test program was Mr. George Nies of Union Carbide Corp.

TEST PROCEDURE AND RESULTS

GENERAL

VISUAL RECEIVING INSPECTION

A visual examination was performed upon receipt of the test item at Dayton T. Brown, Inc. Testing Laboratory. The test item was visually examined for any evidence of damage due to shipping or mishandling. In addition, any apparent abnormality observed in the condition of the test item was noted.

VISUAL RECEIVING INSPECTION (Continued)

The following anomalies were noted during the receiving inspection:

1) Minor scratches and tool marks on external surfaces of the unit. Several nicks and gouges were also noted on unit exterior.

2) Viewing the tank from the neck with the thin wall stainless steel tubing ports on the starboard side, several gouges were noted approximately in line with tubing ports on the circumference of the forward weld.

TEST INSTRUMENTATION

VIBRATION TEST INSTRUMENTATION

One (1) tri-axial response accelerometer (referred to as TP1) was mounted on the test fixture adjacent to a unit mounting point and was utilized to monitor the input vibration. A tri-axial accelerometer (referred to as TP2) was mounted on the top of the test tank (reference photo 1 of enclosure 3). The output signals of the control and response accelerometers were recorded on magnetic tape and subsequently played back as X-Y plots of frequency versus acceleration. Data playback was performed utilizing a 20 cycle filter from 20 to 150 cps.

TEST PHOTOGRAPH

A typical test setup photograph showing the test item during testing is presented in enclosure 3.

TEST TOLERANCE

SINUSOIDAL VIBRATION TOLERANCE

Frequency - 2%
Amplitude - 10%

TEST METHODS AND REQUIREMENTS

The test item was subjected to the following testing as requested in reference (a) and in accordance with reference (c).

VIBRATION TEST

The test unit, as shown in photo 1 of enclosure 3, was subjected to the following vibration in the fore and aft test plane.

Test Frequency	Applied Input
20 cps to 150 cps	$\pm$ 1.0g
20 cps to 150 cps	$\pm$ 10.0g

VIBRATION TEST (Continued)

One sweep was performed from 20 cps to 150 cps to 20 cps in fifteen minutes at each of the above two specified inputs.

The vibration input was limited by sensing the crosstalk response values at the tri-axial response (on the unit only) and limiting the input utilizing an Automatic Transfer Control when crosstalk response exceeded ± 1g during the 1.0g scan and ± 4.0g during the 10.0g scan.

VIBRATION TEST RESULTS

At the completion of the 1.0g scan, a visual inspection of the unit at this time revealed a crack in the weld on the brace of the aft ring. The crack was approximately 3/8 of an inch in length. The cognizant Union Carbide representative stated that he was aware of this crack prior to testing and was apparently overlooked during the initial receiving inspection.

There were no further anomalies noted during the remainder of testing.

Reference enclosure 2, pages 1 thru 6, for test graphs recorded during the 1.0g scan and pages 7 thru 12 for graphs recorded during the 10.0g scan.

If you require any additional information, please do not hesitate to contact the undersigned.

Very truly yours,

DAYTON T. BROWN, INC.

H. D. Pomponio

H. D. POMPONIO
TECHNICAL ADMINISTRATOR

RB/jma

This report contains:
3 Pages
Enclosures: (1) Test Equipment List - 2 Pages
(2) Vibration Test Graphs - 12 Pages
(3) Typical Test Setup Photograph - 1 Photo

ENCLOSURE 1

Test Equipment List
2 Pages

DAYTON T. BROWN INC.

TEST EQUIPMENT

ITEM	MANUFACTURER	MODEL	S/N	CAL. PERIOD	DATE OF LAST CAL.	ACCURACY	REMARKS
Timer	Dimco Grey	165	47-122	6 months	5 Feb 70	± 2%	
Vibration Exciter	M.B. Electronics	C-210	222	TRANSFER INSTRUMENT			
Power Amplifier	M.B. Electronics	T999A	112	TRANSFER INSTRUMENT			
TRMS Voltmeter	Balantine	321	403	3 months	21 Jan 70	± 3% F.S.	
Electronic Counter	Beckman	7350AR	1910	1 month	1 Apr 70	± 1 Count	
Wide Range Oscillator	Hewlett Packard	200CDR	605-61685	1 month	4 Feb 70	± 2%	
Filter	S.K.L.	308A	379	6 months	19 Nov 69	Data	
X-Y Recorder	F.L. Moseley	7035A	604-00122	3 months	30 Mar 70	± 1%	
X-Y Recorder	F.L. Moseley	7035B	845-02958	3 months	4 Feb 70	± 1%	
Log Converter	F.L. Moseley	7561A	531-00189	3 months	21 Mar 70	0.7db Full Scale 20 cps to 10kc	
Log Converter	F.L. Moseley	60D	531-01597	3 months	17 Feb 70	0.7db Full Scale 20 cps to 10kc	
Magnetic Tape Recorder	Sandborn	3914A	104	6 months	4 Nov 69	±1% repro. 3db freq. response	

DAYTON T. BROWN INC.

TEST EQUIPMENT

ITEM	MANUFACTURER	MODEL	S/N	CAL. PERIOD	DATE OF LAST CAL.	ACCURACY	REMARKS
Automatic Vibration Control	Bruel & Kjaer	N576	236	3 months	10 Apr 70	4% Meter 1% Freq.	
Amplitude Transfer Control	Unholtz Dickie	ATC-6	104	TRANSFER INSTRUMENT			
Charge Amplifier	Unholtz Dickie	8PMCV	50-56	3 months	20 Mar 70	5%	
Charge Amplifier	Unholtz Dickie	8PMCV	50-57	3 months	21 Mar 70	5%	
Charge Amplifier Power Supply	Unholtz Dickie	CV608 RMG-6	183	TRANSFER INSTRUMENT			
Charge Amplifier	Unholtz Dickie	8PMCV	50-61	3 months	10 Mar 70	5%	
Charge Amplifier	Unholtz Dickie	8PMCV	50-65	3 months	10 Mar 70	5%	
Charge Amplifier	Unholtz Dickie	8PMCV	50-68	3 months	10 Mar 70	5%	
Charge Amplifier	Unholtz Dickie	8PMCV	50-64	3 months	10 Mar 70	5%	
Accelerometer	Endevco	2228B	NA27	3 months	2 Apr 70	5%	
Accelerometer	Endevco	2228B	PB72	3 months	16 Feb 70	5%	
Accelerometer	Endevco	2228C	TB70	3 months	19 Mar 70	5%	

LAB FORM [illegible]

ENCLOSURE 2

Vibration Test Graphs

12 Pages

46 7522

Vibrator Operator: W. CLENDINNEN

Plotted by: F. Eheman

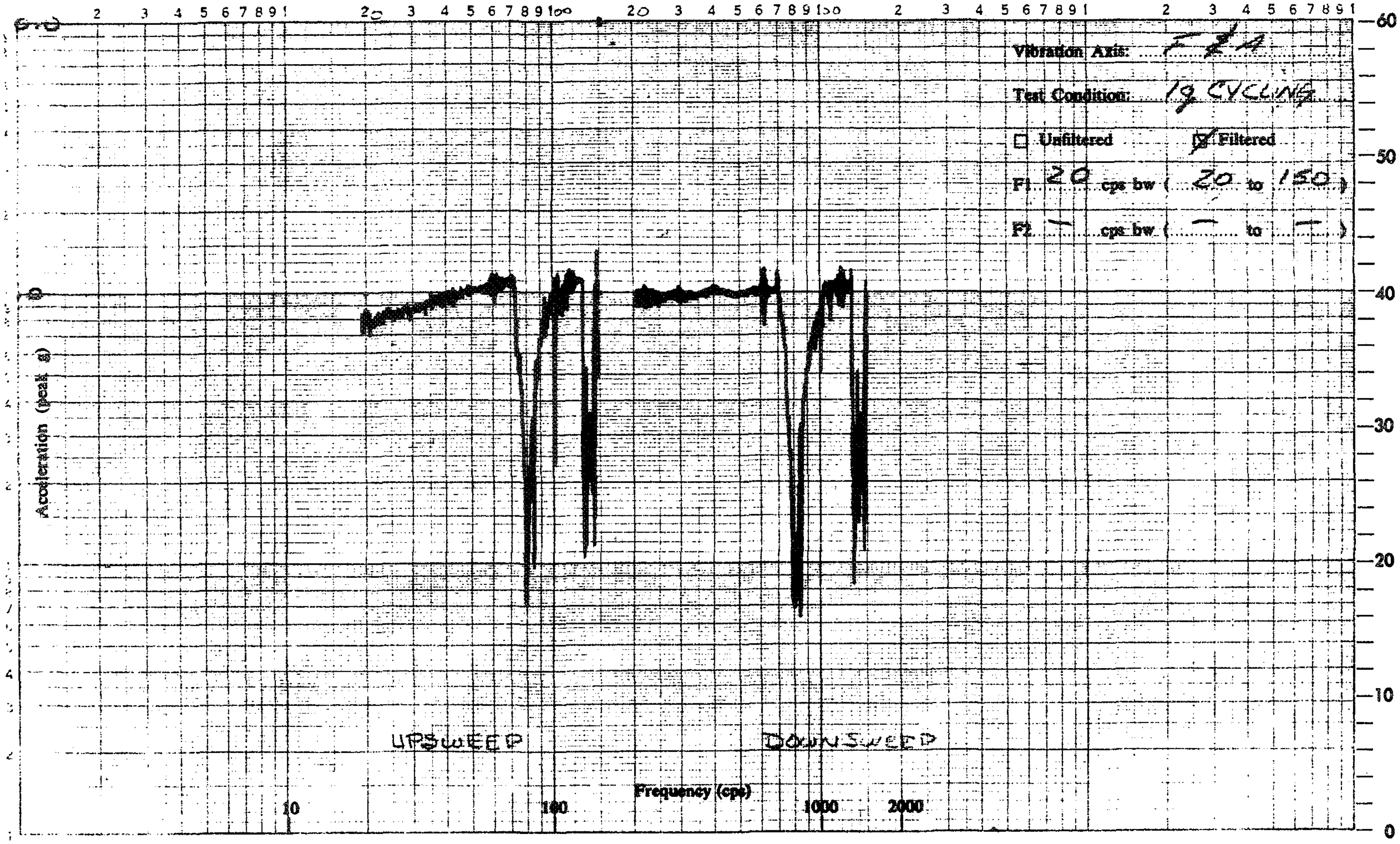
Checked by:

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☒



Graph Number:

Pickup Serial Number: TE 70?

Pickup Sensitivity: 25.0 mv peak / g peak

Job Number: 8565

Pickup Location:

Sweep Speed: 0.4 oct/minute

Date: 15 APR 70

Pickup Sensing Axis:

☐ Live ☐ Tape I.D. Reel 1

Time: 1330

46 7522

Vibrator Operator: W. CLENDINNEN

Plotted by: D. PENDRICH

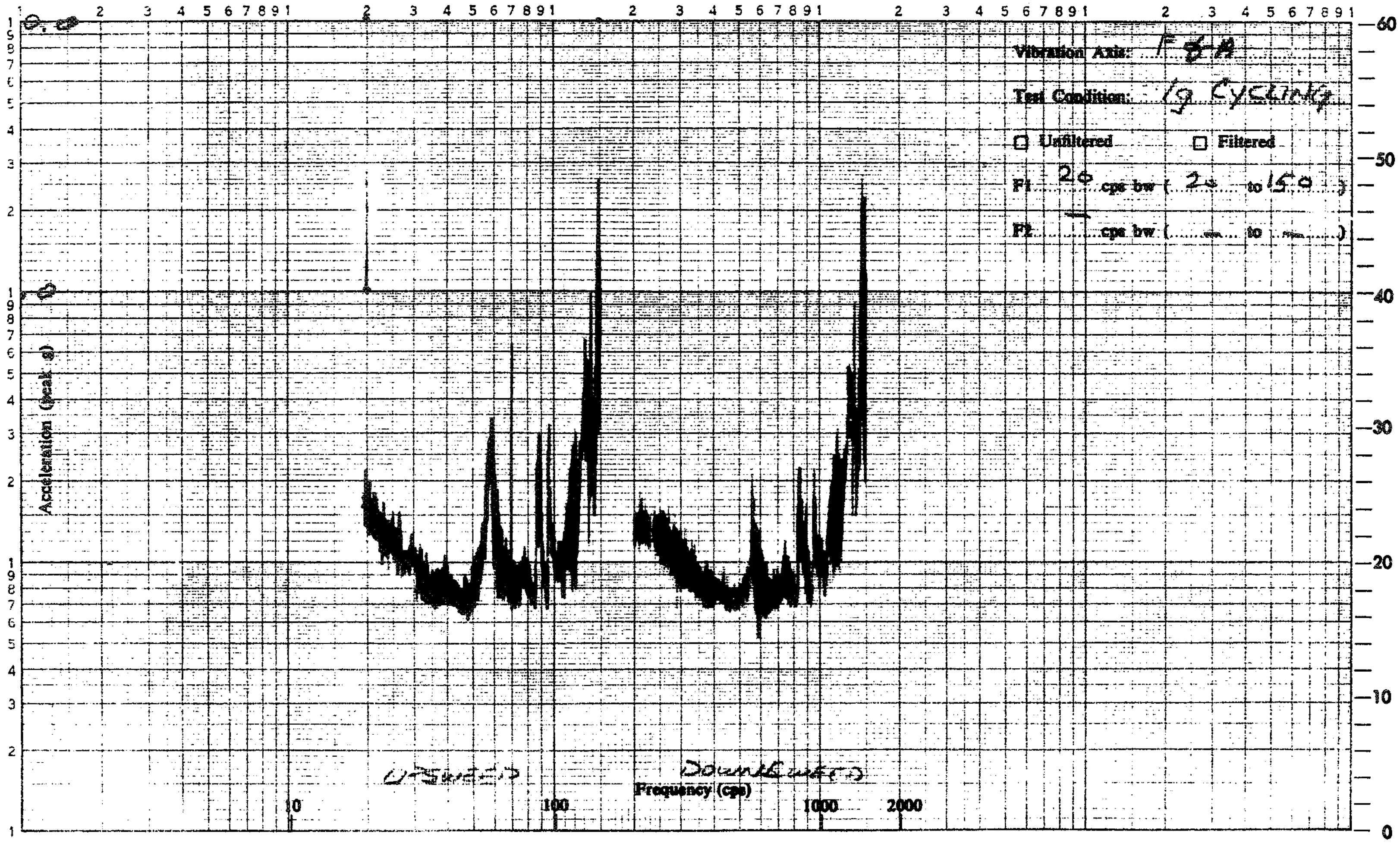
Checked by:

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☐



Graph Number:

Pickup Serial Number: TB70X

Pickup Location: TP1

Pickup Sensing Axis: TRANS

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 1

Job Number: 8565

Date: 15 APRIL 70

Time: 1330

46 7522

Vibrator Operator: W. CLENDINNEN

Plotted by: D. PENDRICK

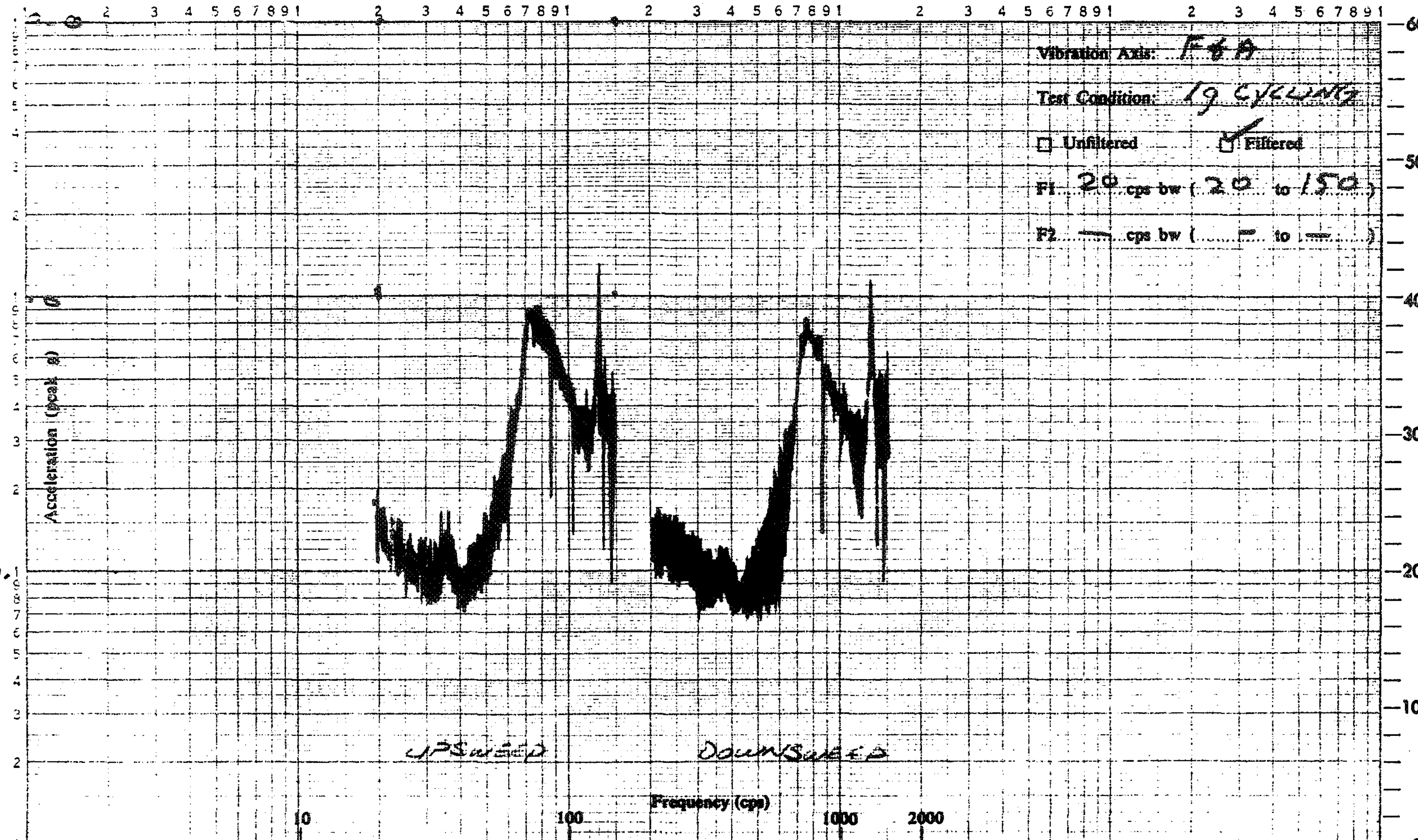
Checked by:

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Pickup Serial Number: TB 70 Y

Pickup Location: TPI

Pickup Sensing Axis: VERT

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 1

Job Number: 8565

Date: 15 APRIL 70

Time: 1330

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W. CLENDINNEN

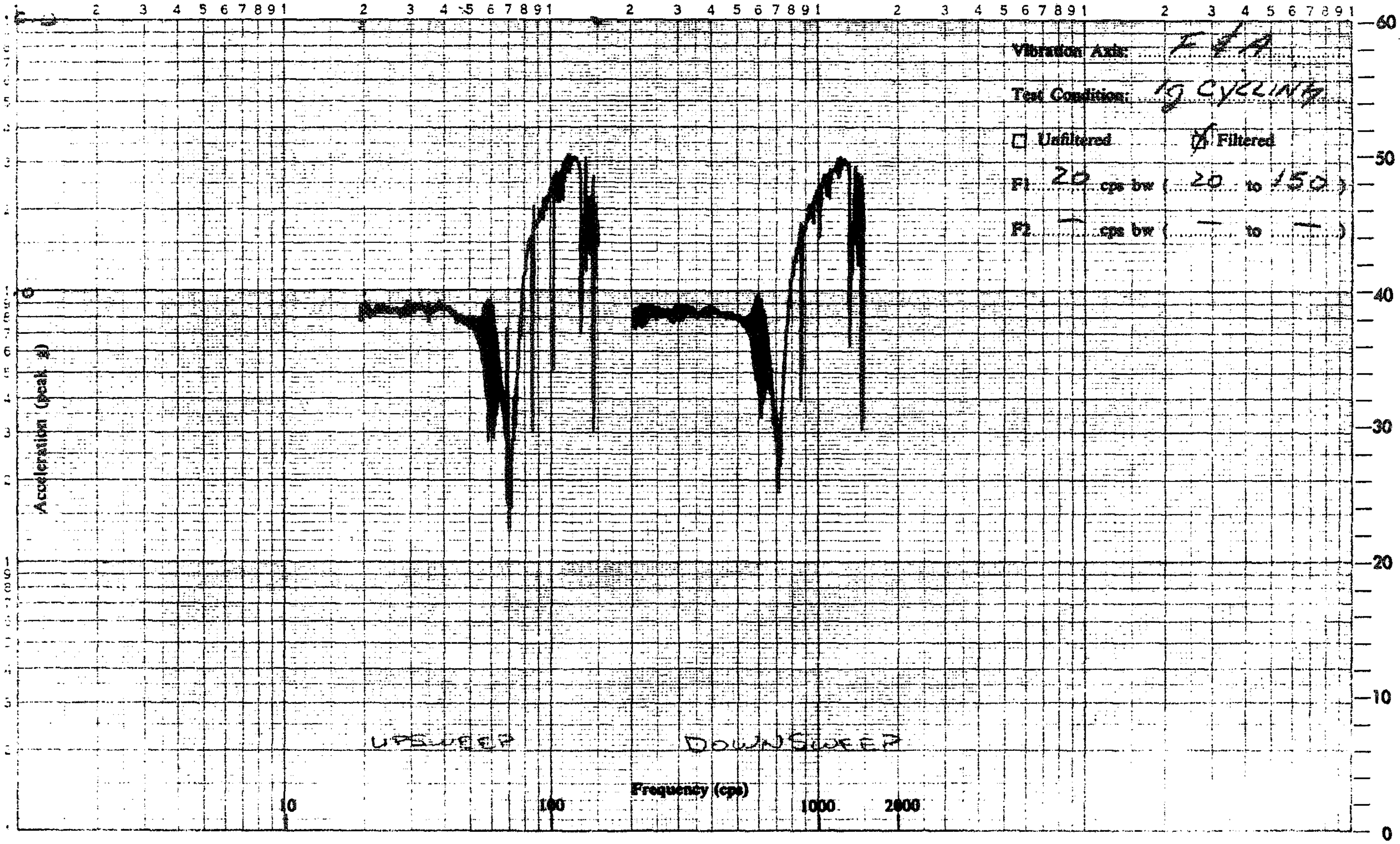
Plotted by: F. Ekman

Checked by:

Test Item: TAI

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Graph Number:

Pickup Serial Number: NA-??

Pickup Location: TP2

Pickup Sensing Axis: F/A

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☒ Tape I.D. Reel 1

Job Number: 9565

Date:

Time:

Lab Form D-24

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W. Clendennen

Plotted by: [illegible]

Checked by:

Test Item: Tank

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☐

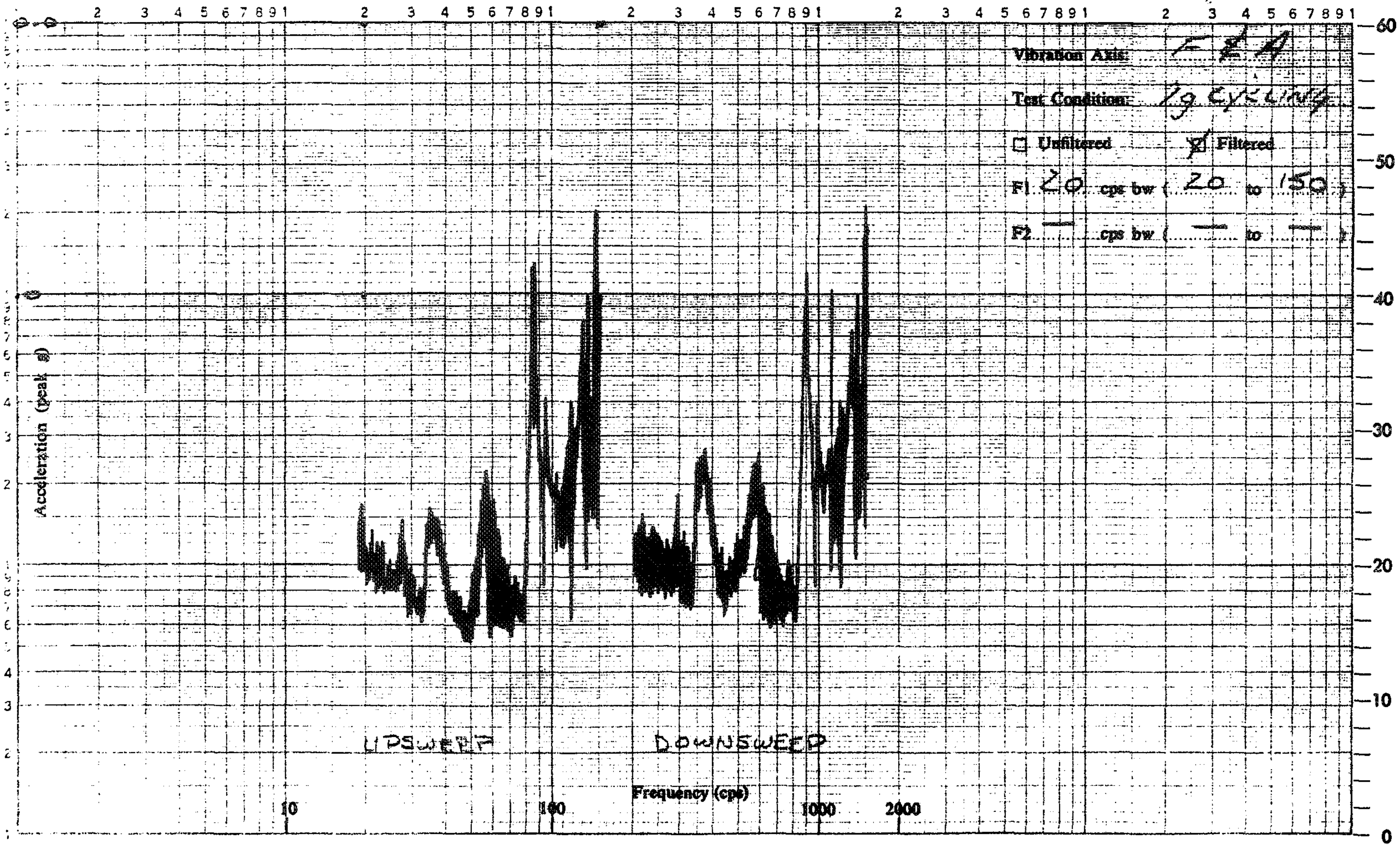
Vibration Axis: F & A

Test Condition: 1g Cycling

☐ Unfiltered ☒ Filtered

F1 20 cps bw (20 to 150)

F2 — cps bw (— to —)



Graph Number:

Pickup Serial Number: NA27X

Pickup Location: [illegible]

Pickup Sensing Axis: [illegible]

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☒ Tape I.D. Reel 1

Job Number: 8565

Date: 5 Jun 70

Time: 1330

Vibrator Operator: W. CLENDINNEN

Plotted by: D. PENDZICK

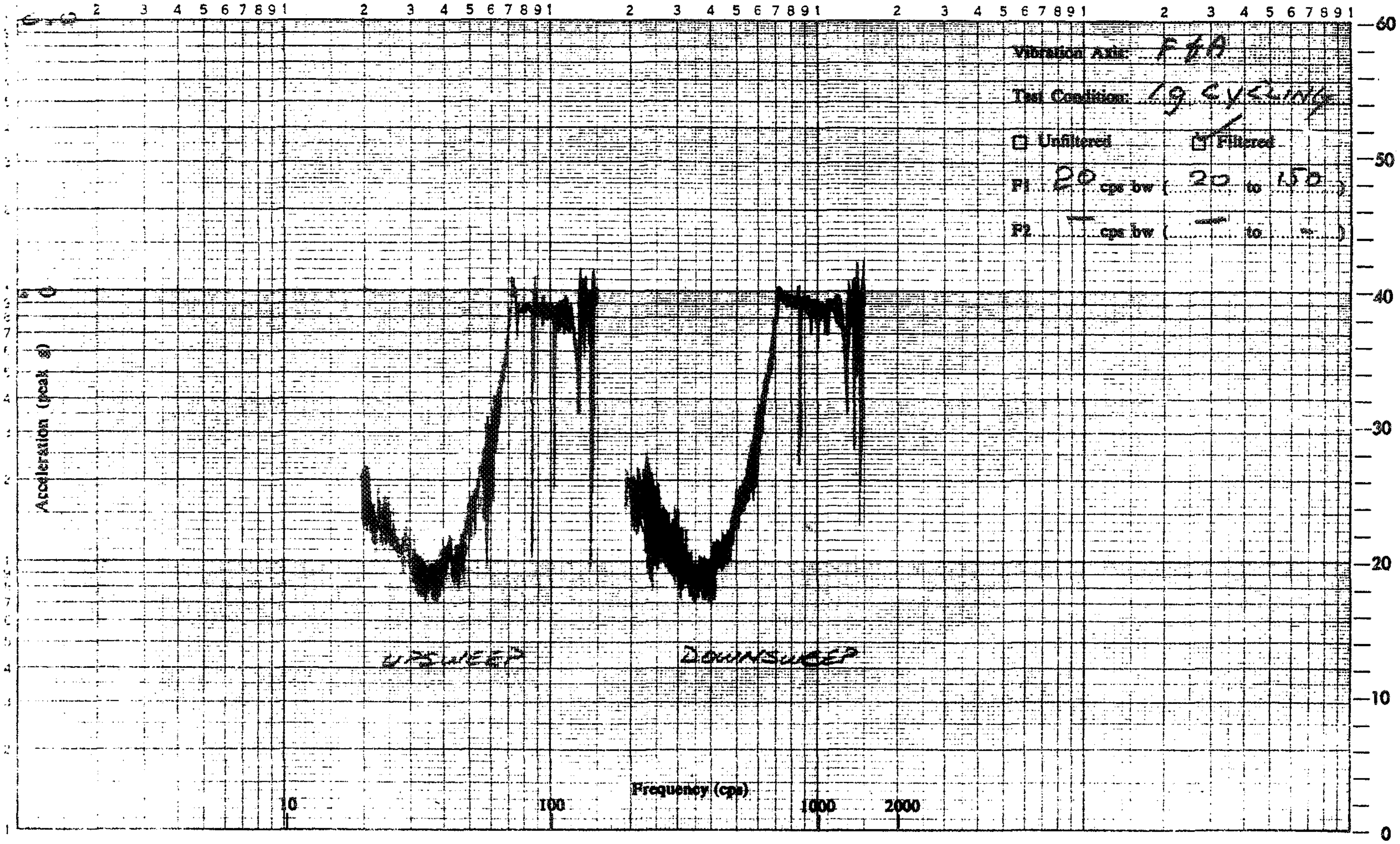
Checked by: ....

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Graph Number: ....

Pickup Serial Number: NA27Z

Pickup Location: TP2

Pickup Sensing Axis: VERTICAL

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. .... Reel 1

Job Number: 8565

Date: 15 APRIL 70

Time: 1330

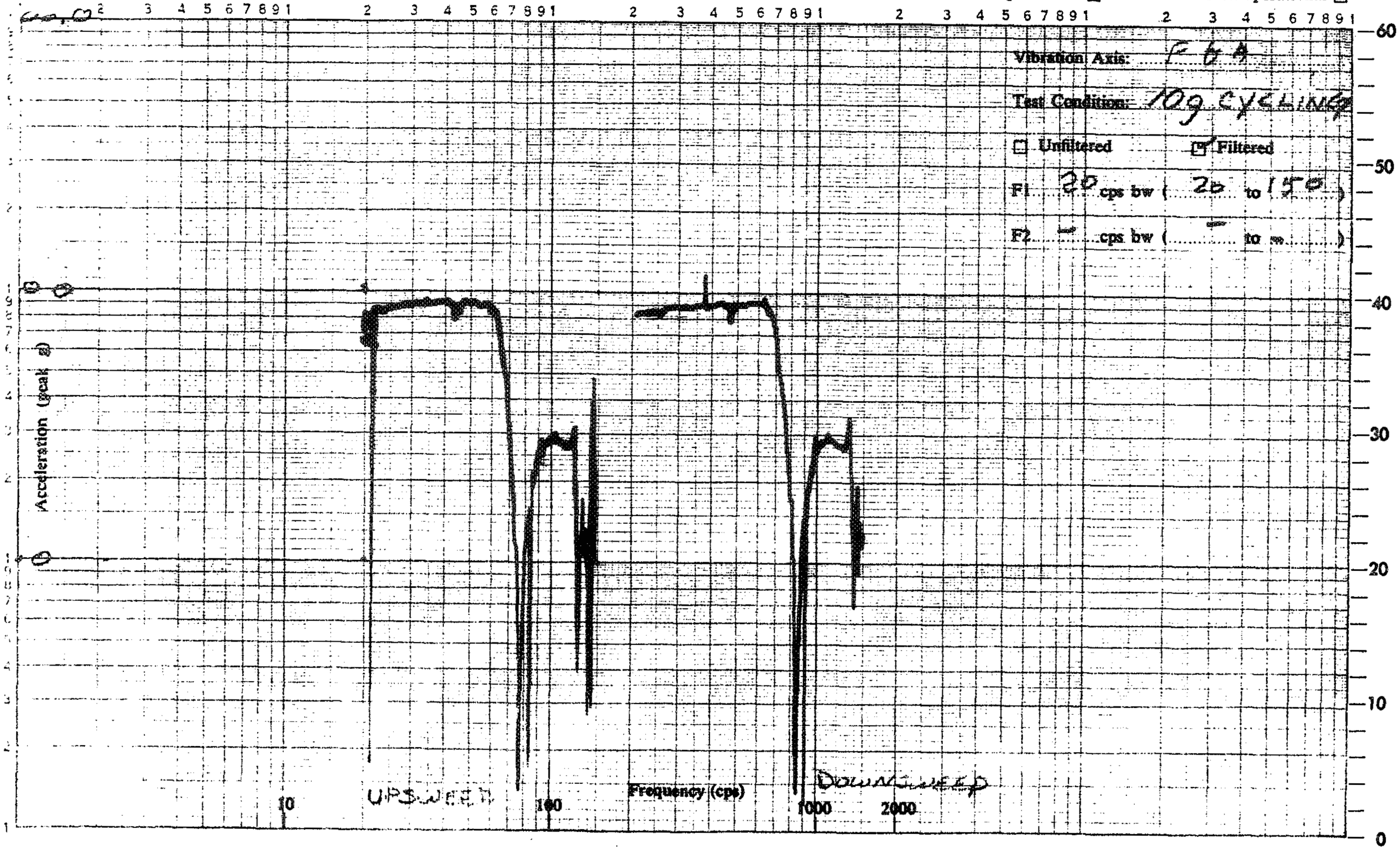
46 7522

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W. CLENDINNEN
Plotted by: D. PENDZKA
Checked by:

Test Item: TANK
Serial Number(s): 629
Unit: Operational ☐ Non-operational ☐

Vibration Axis: F & A
Test Condition: 10g CYCLING
☐ Unfiltered ☑ Filtered
F1 20 cps bw (20 to 150)
F2 — cps bw (— to ∞)



Graph Number:

Pickup Serial Number: TG70Z
Pickup Location:
Pickup Sensing Axis: F & A

Pickup Sensitivity: 25.0 mv peak / g peak
Sweep Speed: 0.4 oct/minute
☐ Live ☑ Tape I.D. Reel 1

Job Number: 8565
Date: 15 APRIL 70
Time: 1541

Vibrator Operator: W. CLENDINNEN

Plotted by: D. PENDRICK

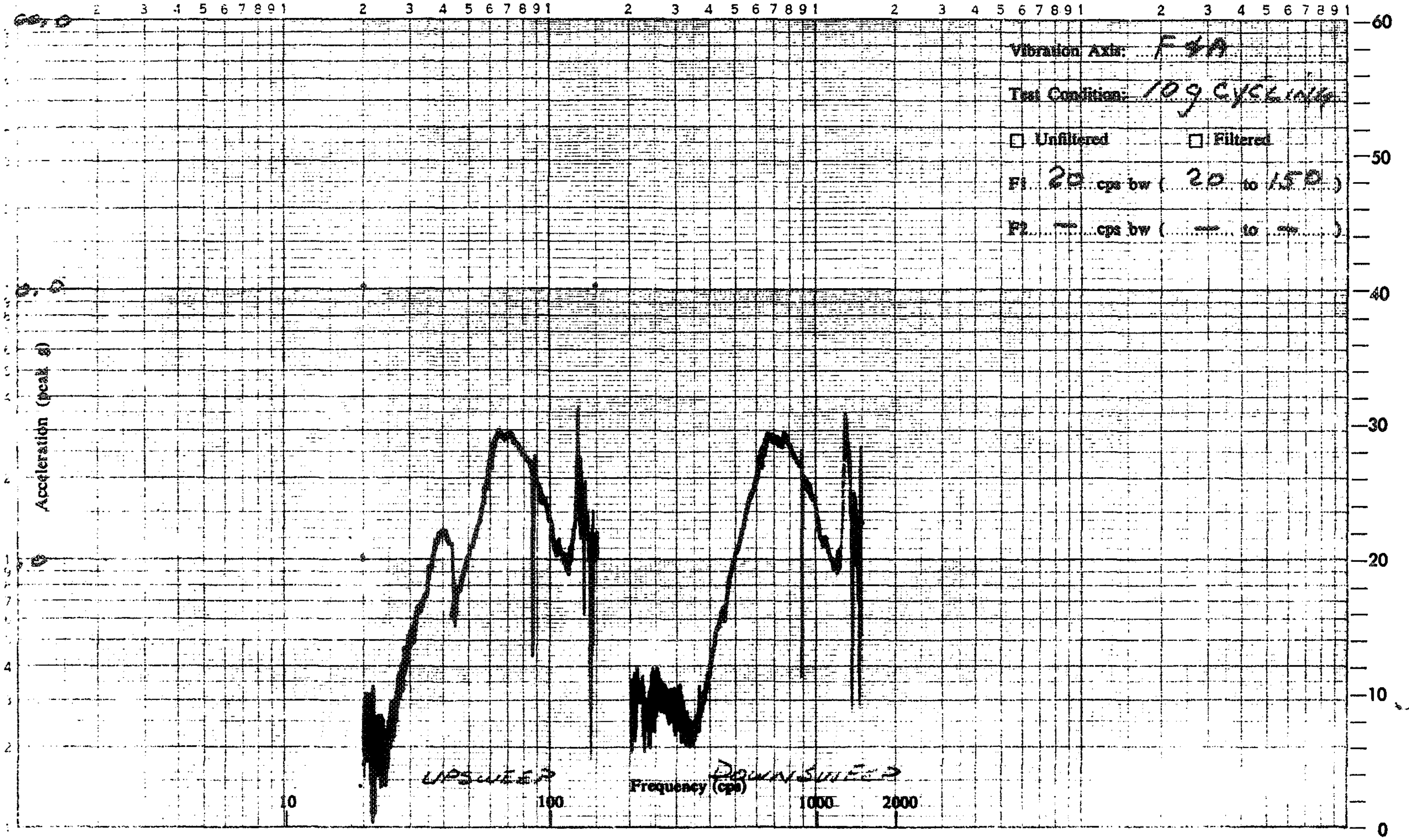
Checked by:

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Graph Number:

Pickup Serial Number: TB70Y

Pickup Location: TP1

Pickup Sensing Axis: VERTICAL

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 1

Job Number: 8565

Date: 15 APRIL 70

Time: 1541

Lab Form D-24

46 7522

Vibrator Operator: W. CLENDINNEN

Plotted by: D. PENDRICK

Checked by:

DAYTON T. BROWN INC.
Testing Laboratories

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑

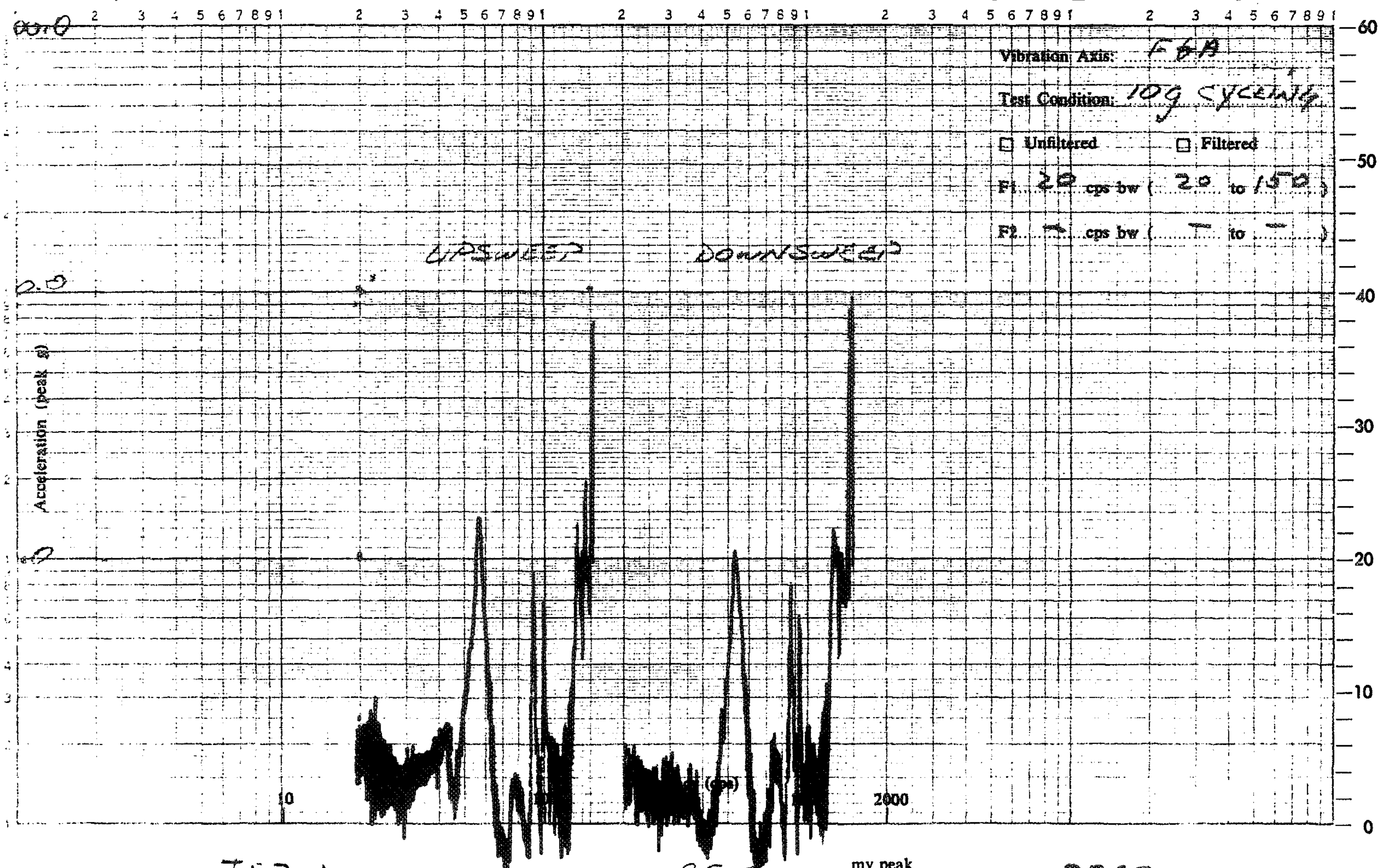
Vibration Axis: F&A

Test Condition: 10g CYCLING

☐ Unfiltered ☐ Filtered

F1 20 cps bw (20 to 150)

F2 — cps bw (— to —)



Graph Number:

Pickup Serial Number: TB70X

Pickup Location: TC1

Pickup Sensing Axis: TANK

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 4

Job Number: 8565

Date: 15 APRIL70

Time: 1541

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W C Lenowmen

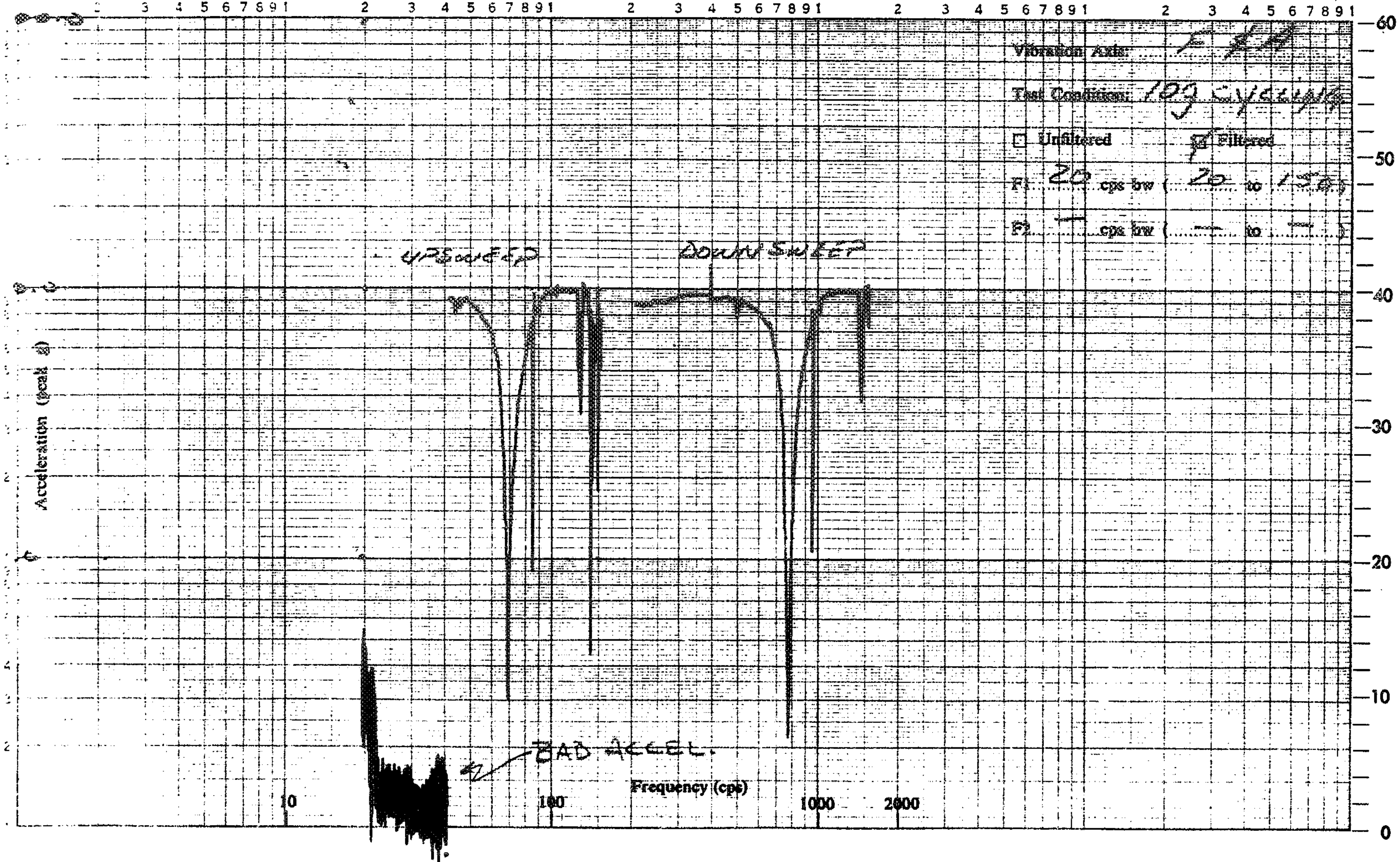
Plotted by: [illegible signature]

Checked by:

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☒



Graph Number: 10

Pickup Serial Number: NA27Y

Pickup Location: TP2

Pickup Sensing Axis: Fore & Aft

Pickup Sensitivity: 250 mv peak / g peak

Sweep Speed: 0.1 oct/minute

☐ Live ☐ Tape I.D. Reel

Job Number: 8565

Date: 15 APR 70

Time: 1541

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W. CLENDINNEN

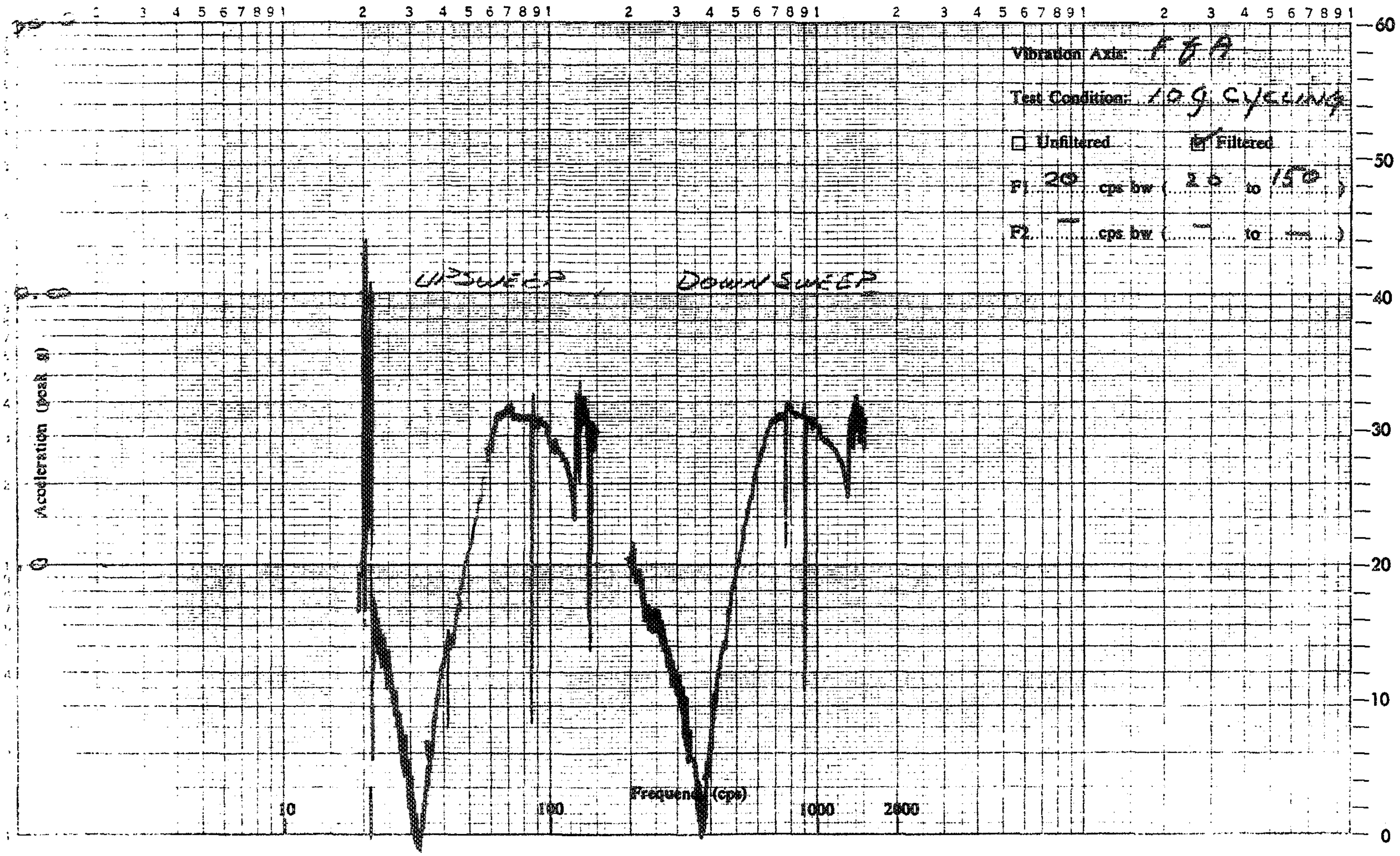
Plotted by: D. PENDZICK

Checked by:

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Pickup Serial Number: NA27Z

Pickup Location: TP2

Pickup Sensing Axis: VERT

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 1

Job Number: 8565

Date: 15 APRIL 70

Time: 1330

K+E LOGARITHMIC 46 7522
KEUFFEL & ESSER CO.

DAYTON T. BROWN INC.
Testing Laboratories

Vibrator Operator: W. CLENDINNEN

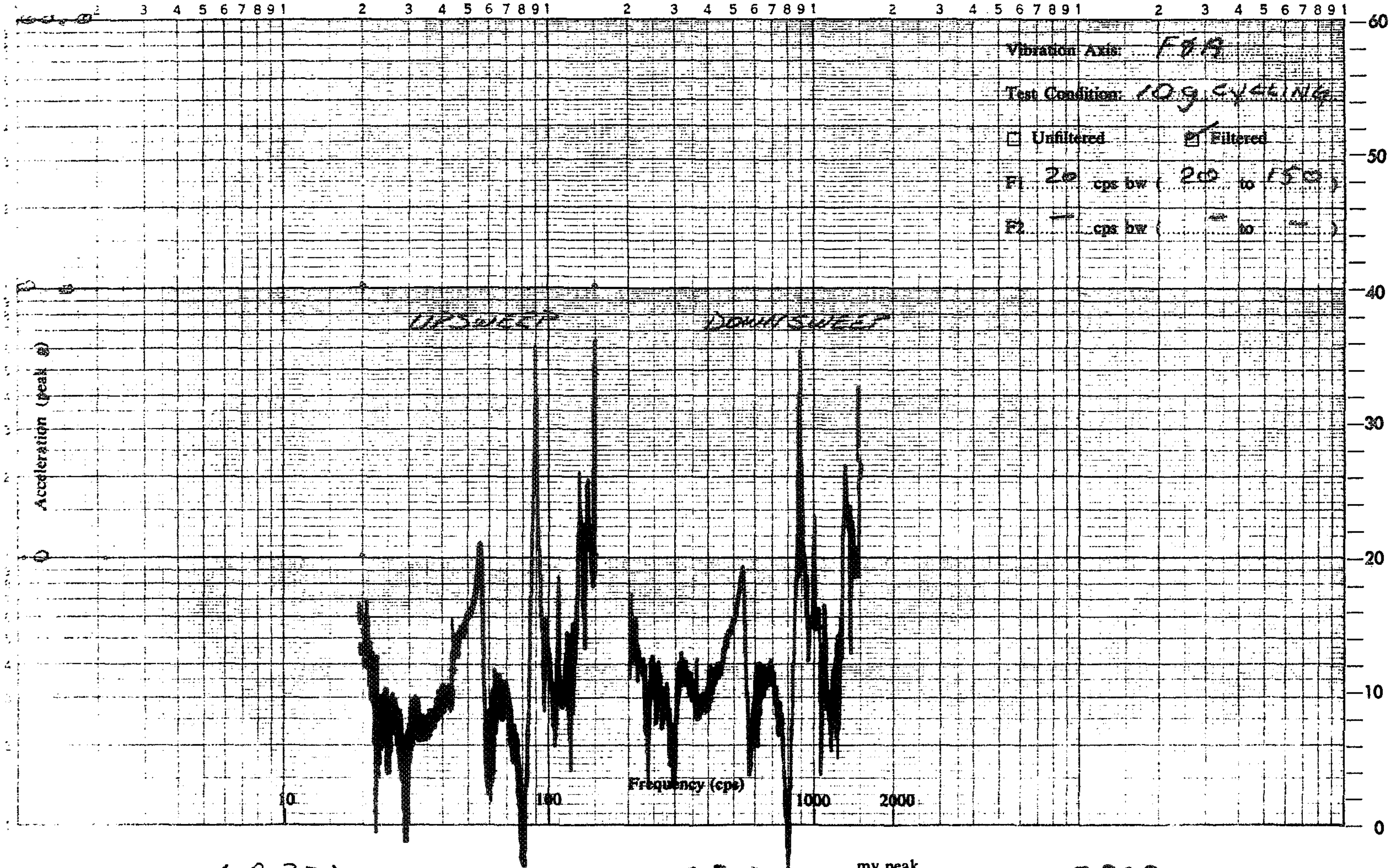
Plotted by: D. PENDRICK

Checked by:

Test Item: TANK

Serial Number(s): 629

Unit: Operational ☐ Non-operational ☑



Graph Number:

Pickup Serial Number: NA 27X

Pickup Location: TP2

Pickup Sensing Axis: TRANS.

Pickup Sensitivity: 25.0 mv peak / g peak

Sweep Speed: 0.4 oct/minute

☐ Live ☑ Tape I.D. Reel 1

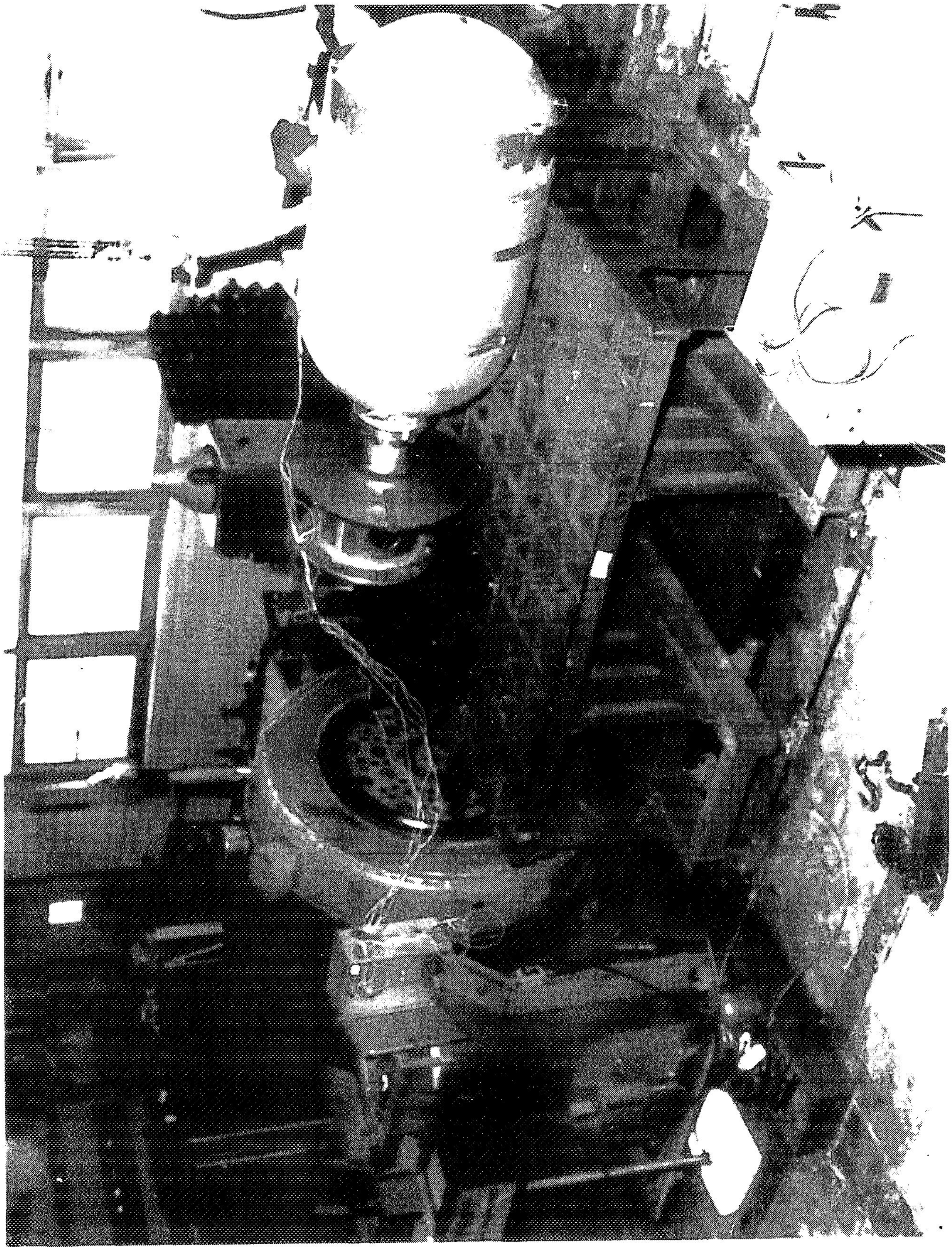
Job Number: 8565

Date: 15 APRIL 70

Time: 1541

ENCLOSURE 3

One (1) Photograph



DAYTON T. BROWN, INC. - TESTING LABORATORIES
TESTED FOR: UNION CARBIDE CORPORATION ITEM: TANK
MFG: UNION CARBIDE CORPORATION MODEL: TEST S/N: 629
TYPICAL VIBRATION TEST SETUP; TEST UNIT SHOWN SECURED
TO VIBRATION EXCITER IN THE FORE & AFT TEST PLANE
DTR JOB NO.: 8565 PHOTO FILE #: 4287 DATE: 15 APRIL 1970
DTR03R70-0587 ENCLOSURE: 3 PHOTOGRAPH 1 of 1

APPENDIX 8

TEST PLAN

YOR 1327

TITLE: Self Evacuating Multilayer Insulation Environmental Testing

TEST PROGRAM NUMBER:

PLUM BROOK TEST SITE: "K"

SPONSORING DIVISION: Chemical Rocket Division

CLEVELAND PROJECT ENGINEER: J. R. Faddoul

PLUM BROOK PROJECT ENGINEER: J. E. Cairelli

APPROVED:

I. A. Johnsen
Chief, Chemical Rocket Division

G. Hennings
Chief, Rocket Systems Division

I. OPERATIONS PLAN

A. Introduction - A self evacuating multilayer insulation (SEMI) system is being investigated for use on cryogenic spacecraft propellant tanks. Previous efforts have demonstrated the thermal effectiveness of the system on cylindrical tankage. The current effort will determine the system performance when applied to practical cryogenic tankage (simulated Centaur tank) and subjected to a typical launch environment. The launch environment will be provided by the Goddard Space Flight Center Launch Phase Simulator. The appropriate thermal performance tests, which are the subject of this document, will be performed at Lewis Research Center, Plum Brook - "K-site".

B. Objective - The thermal evaluation of the Self Evacuating Multilayer Insulation (SEMI) system at Plum Brook consists of three tests. The objective of the first test is to determine the thermal performance of the SEMI system in the as received condition. The objective of the second test is to determine any system degradation caused by long term exposure to a normal atmospheric environment and to provide baseline system performance data prior to subjecting the SEMI system to a simulated launch load profile. The objective of the third thermal test is to determine the system thermal performance degradation due to the launch environment.

C. Description of Research Package - The research package is illustrated by Figure 49. Three basic items comprise the package. Item 1 is the insulated test tank. Item 2 is the isothermal shroud (currently being used with LH_2 coolant under R3119). And Item 3 is the cold guard to prevent spurious heat leaks from entering the test tank. Because of the necessity to cool the SEMI panels prior to evacuating the chamber, it will be necessary to prevent the vacuum chamber atmosphere from condensing on the cold guard. The He system, schematically illustrated in Figure 49, is accordingly provided to purge the region surrounding the cold guard. (The chamber cannot be purged with He gas since this would seriously affect the performance of the SEMI panels).

It is expected that the boiloff from the test tank during this initial cooldown phase will be approximately 1000 BTU/hr (5#/hr or 1000 SCFH). During the time when the chamber is evactuated, the figure is expected to be a maximum of 200 SCFH.

D. Facility to be Used - The thermal testing of the SEMI system will be performed in "K" site Plum Brook. No other facilities will be required. Due to the nature of the SEMI panels, the chamber pressure cannot be reduced until a vacuum has been achieved in the panels. Thus, the panels must be cold (condensing the CO_2 within)

D. (continued)
before chamber pumpdown is initiated. In addition, the permeable nature of the panel materials precludes having a pure helium environment in the K-site chamber. Thus, nitrogen gas must be used to provide an inert environment during cooling of the SEMI panels. To prevent condensation and freezing of the nitrogen gas, the test tank LH_2 fill line must be insulated. In addition, the cold guard will be purged with GHe since insulating that area would be extremely difficult. Also, to provide a controlled environment for the SEMI system during test, the 8' diameter shroud, fabricated by ADL and currently being used with LH_2 coolant under R3119, will be used with 70°F water.

The SEMI system testing in K-site thus dictates three minor modifications to the facility. A GN_2 chamber purge must be provided, a GHe purge and external vent capability must be provided to the enclosed region exterior to the cold guard, and a constant temperature water source must be provided to the isothermal shroud.

Data acquisition will require no new equipment. The 30 KC SEL Multiplexer System will be more than adequate. A maximum of 150 chanels will be required.

E. Test Setup - The test setup is shown in Figure 50. Only LH_2 will be used for filling the test and guard tanks. The LN_2 lines will not be used but are shown because they are presently part of the fill system. The pressures in the guard and test tank will be controlled independently by separate "closed-loop back pressure control systems."

Boiloff from the tank will be determined by the use of a boiloff meter. The expected flow rate (200 SCFH) is well within the capability of the meters currently being used under R3119.

The shroud will be used to provide a constant temperature environment around the test tank. Water, at a temperature of 70°F ± 10°F will be supplied to the coils on the cylindrical portion of the shroud and to the lower end cap. Water will not be circulated through the upper cap of the shroud.

F. Instrumentation - The position and types of instrumentation on the test tank is shown in Figure 51. All instrumentation types and locations are given in Table I.

G. Test Conditions - The test series consists of three separate tests which are to be run under identical conditions. Each test will consist solely of obtaining the steady state heat flux to the insulated test tank with the K-site environmental chamber evacuated and 70°F ± 10°F water flowing through the sides and bottom of the 8' shroud.

H. General Test Procedure - The test procedure for each test will be as follows:

1. Check and/or zero all gas, liquid and electrical connections and equipment.

2. Start GN_2 environmental chamber purge.

3. Start $70^{o}F \pm 10^{o}F$ water flow through shroud.

4. Start GHe cold guard purge.

5. Establish LH_2 flow into test tank.

6. Monitor and record appropriate thermocouples and panel vacuum gauges to determine SEMI panel internal pressure.

7. After SEMI panels have achieved pressures less than 0.1 torr., initiate pumpdown of the environmental chamber and the region within the cold guard purge bag. Note: Monitor thermocouple on flange surface to ascertain that temperature is above N_2 liquefaction point. If this temperature is lower than $-310^{o}F$, discontinue LH_2 flow and allow the flange to warm to $-310^{o}F$ prior to environmental chamber and cold guard purge space evacuation.

8. Fill test tank with LH_2.

9. Fill guard tank with LH_2.

10. Top test tank and guard tank as required.

11. Monitor boiloff until steady state is achieved.

12. Retop test and guard tank and monitor boiloff for 4 to 8 hours to assure steady state conditions have been attained.

13. Empty all LH_2 lines and tanks.

14. Prior to significant warm up of the SEMI panels, back fill the environmental chamber with GN_2 and the cold guard purge space with GHe or GN_2.

15. Maintaining one atmosphere of GN_2 in the environmental chamber allow test hardware to warm to ambient temperature.

16. Stop $70^{o}F \pm 10^{o}F$ water flow and remove equipment.

I. Schedule - A detailed schedule is shown in Figure 52.

J. Pretest and Post-test Operations - Prior to the first test, the insulated test tank will be shipped to Plum Brook from Union Carbide/Linde Division. The system will be contained in a transporter (Figure 53) which can be used for all handling operations. To prepare the tank for installation in K-site, the following operations must be performed for each test sequence:

1. Place purge bag loosely around test tank neck tube flange.

2. Clean cold guard flange edges.

3. Set cold guard and gasket in place on top of test tank in transporter.

4. Bolt cold guard to test tank.

5. Apply adhesive (Narmco 7343/7139) to cold guard flange edges and purge bag edges.

6. Pull purge bag into place.

7. Fix wooden clamp rings to both flanges to apply pressure to bond line - allow to cure for 24 hours.

8. Leak check purge bag.

9. Apply a maximum of 6 thermocouples to upper area of test tank surface and neck tube flange.

10. Insulate the LH_2 test tank fill line inside the K-site environmental chamber.

11. Install shroud and connect purge and electrical lines.

Post test operations would be identical but in reverse order. Between the second and third test sequence, two additional operations are required. After the second test, the two vacuum gages installed in the SEMI panels must be removed and replaced with aluminum plugs which will be provided as part of the test package. After the test package is returned to Plum Brook for the third thermal test, those vacuum gages must be reinstalled. The installation and removal of the vacuum gages shall be performed with simultaneous local CO_2 (Coleman grade) purging of the panel to avoid contamination of the cryopump (CO_2) gas within the panels. A mylar film taped to the surface of the panel will be installed over the work area and shall be fully purged before and during the operation.

At the completion of all testing, and with the assistance of Union Carbide personnel, the SEMI panels will be stripped from the test tank, opened, and examined. Any evidence of damage to any of the components will be documented.

II. MANAGEMENT PLAN

The organizational and individual responsibilities for carrying out all elements of the test program are listed below:

Task	Responsible Station	Responsible Division	Responsible Person
A. Design & Fabrication of Special Test Equipment			
1. Design and install GHe purge and vent lines	PB	Rocket Systems	J. Cairelli
2. Procure or supply clamps for purge bag bonding frames	PB	Rocket Systems	J. Cairelli
3. Provide GN_2 purge capability for environmental chamber	PB	Rocket Systems	J. Cairelli
4. Provide constant temperature water supply for shroud	PB	Rocket Systems	J. Cairelli
5. Supply purge bags, bonding frames and adhesive	LeRC	CRD	J. Faddoul
6. Provide fill and drain line assembly	LeRC	CRD	J. Faddoul

Task	Responsible Station	Responsible Division	Responsible Person
B. Test Hardware Assembly			
1. Install 6 thermocouples on outside of test package	PB	Rocket Systems	R. Rettman
2. Apply purge bag to cold guard and leak check	PB	Rocket Systems	J. Cairelli
3. Insulate LH_2 test tank fill line	PB	Rocket Systems	J. Cairelli
4. Remove vacuum gages and reinstall on return from Goddard	PB	Rocket Systems	R. Rettman
5. Instrument fill and drain pipe with LL sensors	PB	Rocket Systems	R. Rettman
C. Installation of Research Package			
Assemble and install test hardware in chamber	PB	Rocket Systems	J. Cairelli
D. Reporting			
Preparation of report summarizing all SEMI test data	LeRC	Chemical Rocket Div.	J. Faddoul
E. Test Operations			
1. Facility preparation	PB	Rocket Systems	J. Cairelli
2. Setup of computer in control room for on-line computation of boiloff gas rates	PB	Rocket Systems	W. Kuchmeier
F. Data Requirements			
1. 30KC SEL multiplexer system. Strip charts and visicorder, and GE Vacuum gage read-out	PB	Rocket Systems	R. Rettman

Task	Responsible Station	Responsible Division	Responsible Person
F. (continued)			
2. Data reduction. Obtain calibrated data and terminal calculations	LeRC	CRD	J. Faddoul

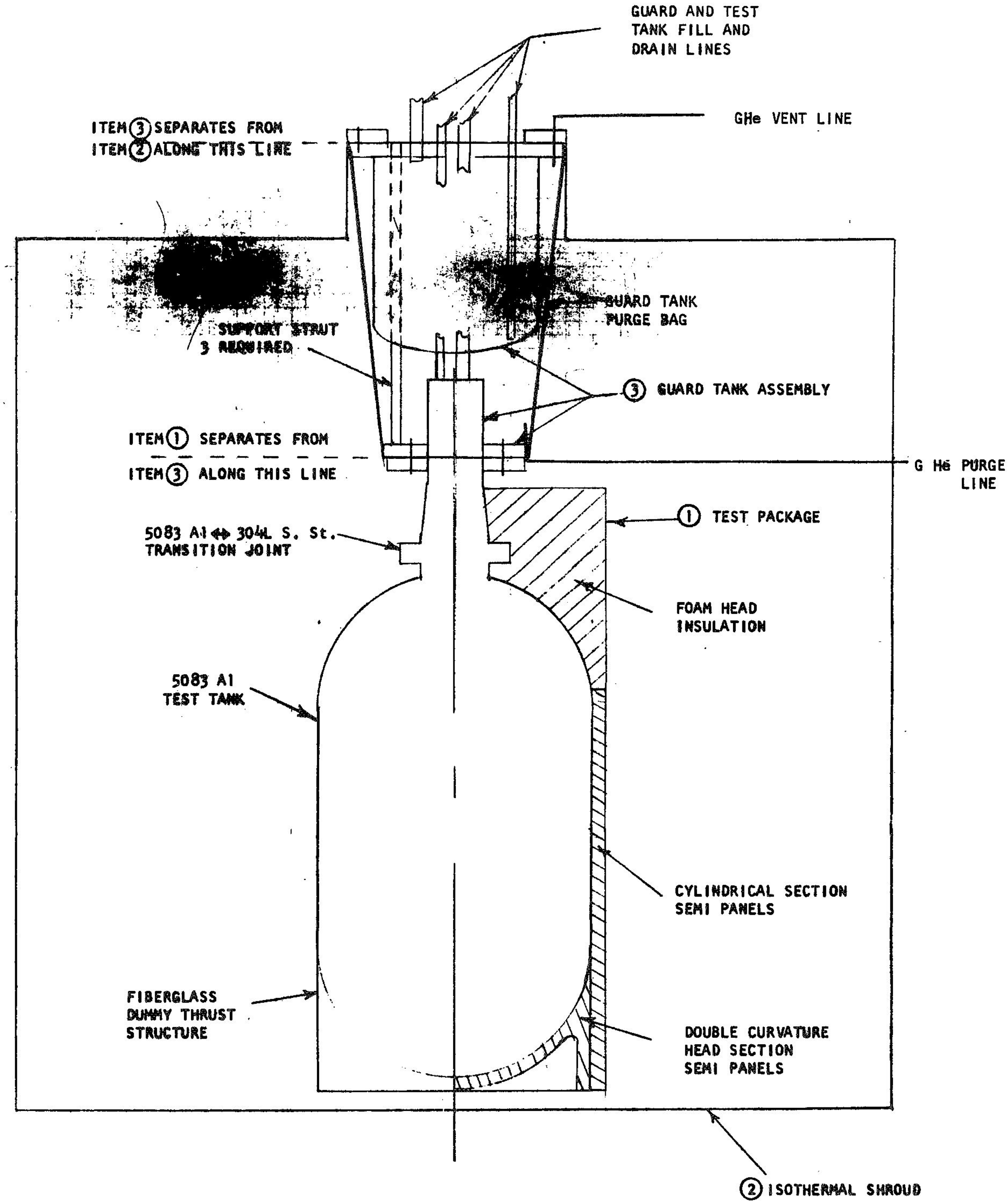
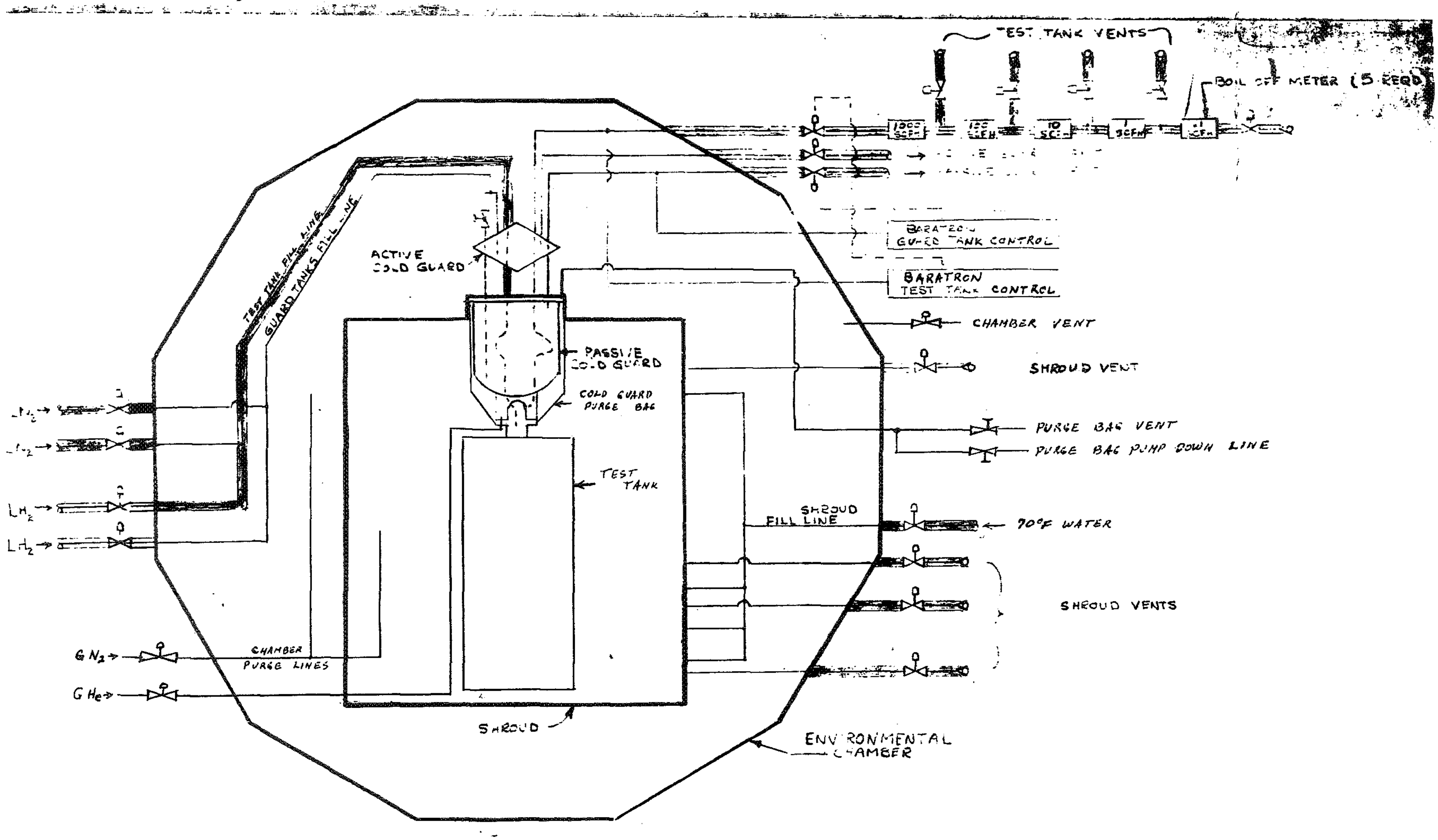
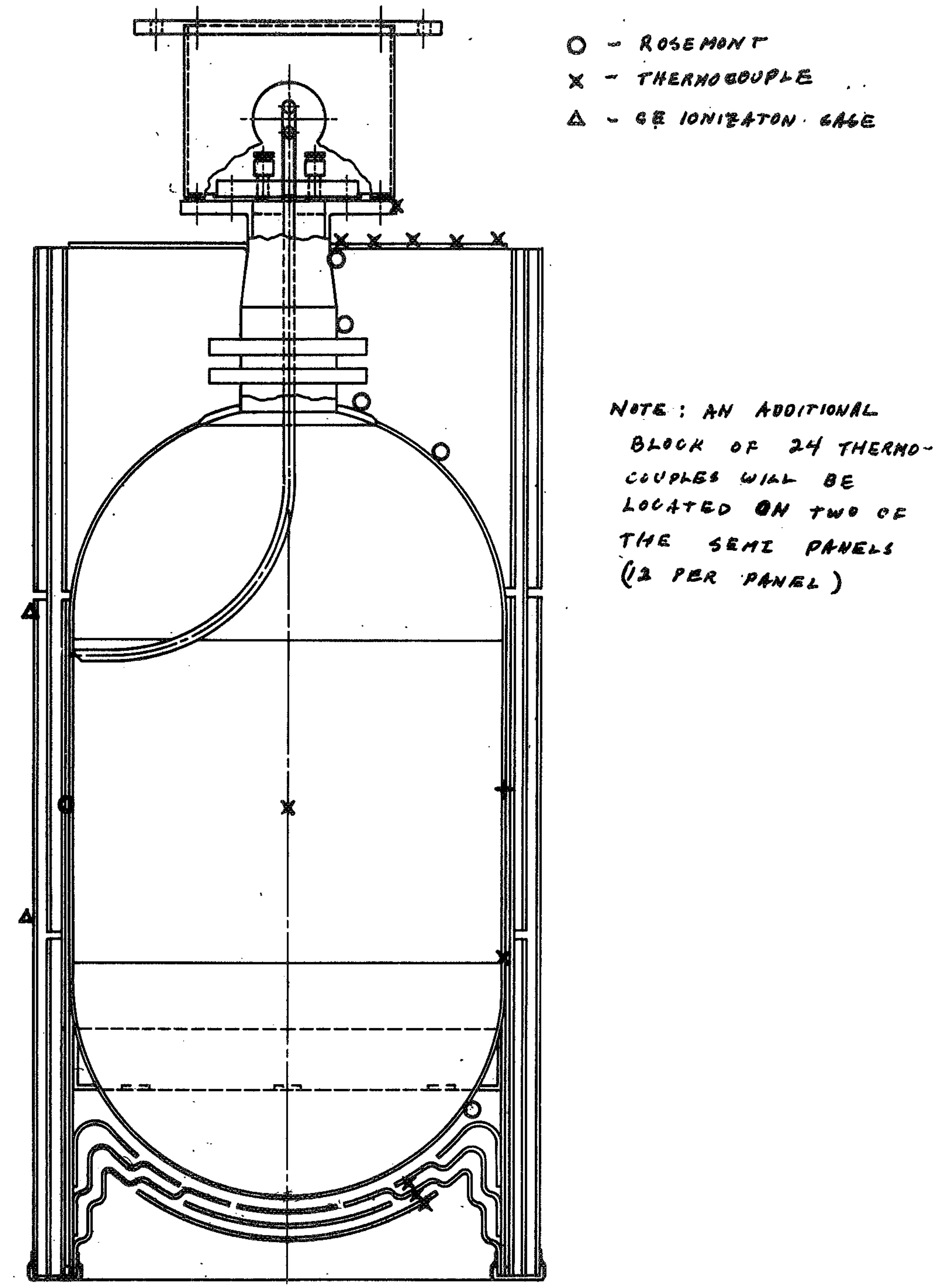


Figure 49 - SCHEMATIC OF TEST EQUIPMENT READY FOR INSTALLATION IN K-SITE



TEST CONFIGURATION INSTALLED IN THE 25' DIA.
ENVIRONMENTAL CHAMBER

FIGURE 50



CD-10682-11

FIGURE 51 - INSTRUMENTATION LOCATIONS

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LEWIS RESEARCH CENTER
CLEVELAND, OHIO 44135

~~CONTRACT~~ PROGRESS SCHEDULE

REPORT FOR MONTH ENDING

1. ~~CONTRACT~~ TITLE: SEMI SYSTEM ENVIRONMENTAL TESTING FIGURE 52

2. CONTRACTOR (Name and address): PLUM BROOK "K-SITE"

3. ~~CONTRACT~~ NO.: YOR 1327

4. APPROVED (Contractor's Project Manager): James R. [signature]

PREPARATION DATE: 3-30-70

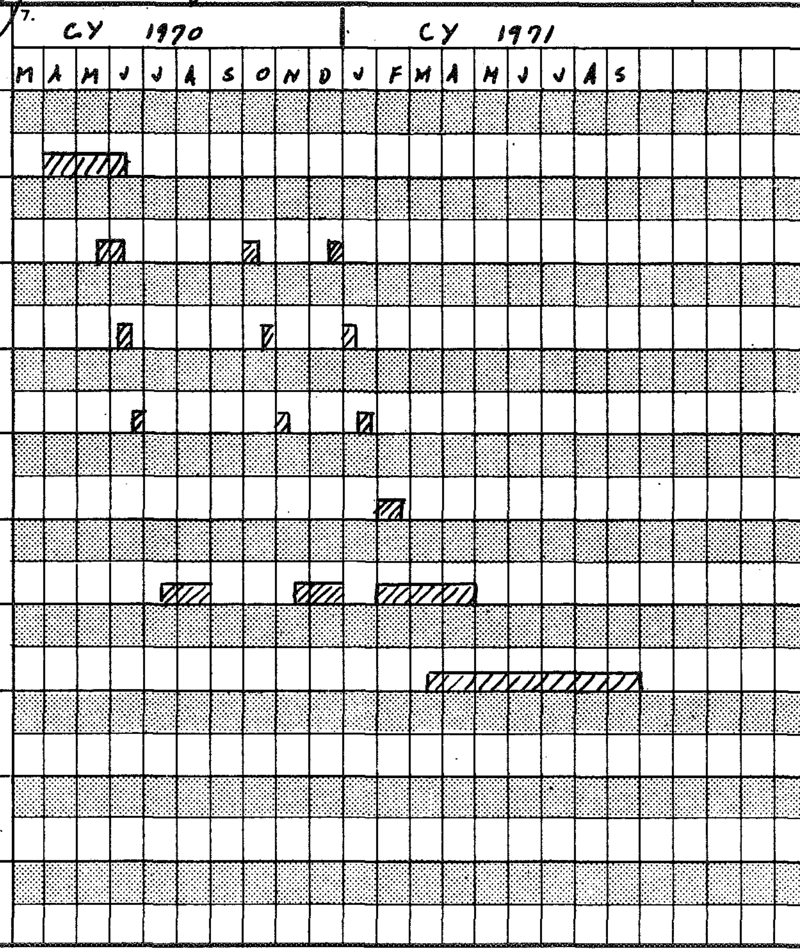
5. NASA APPROVED SCHEDULE DATE:

9. NASA Use Only
a. NASA CODE
b. PROJECT MGR.
c. EVALUATION DATE
d. EXCEPTION CATEGORY
e. f. g.

6. REPORTING CATEGORY

- FACILITY PREPARATION
- TEST PACKAGE ASSEMBLY AND INSTRUMENTATION
- INSTALLATION AND CHECKOUT
- TEST
- DISASSEMBLY AND INSPECTION
- DATA REDUCTION
- REPORT PREPARATION

7.

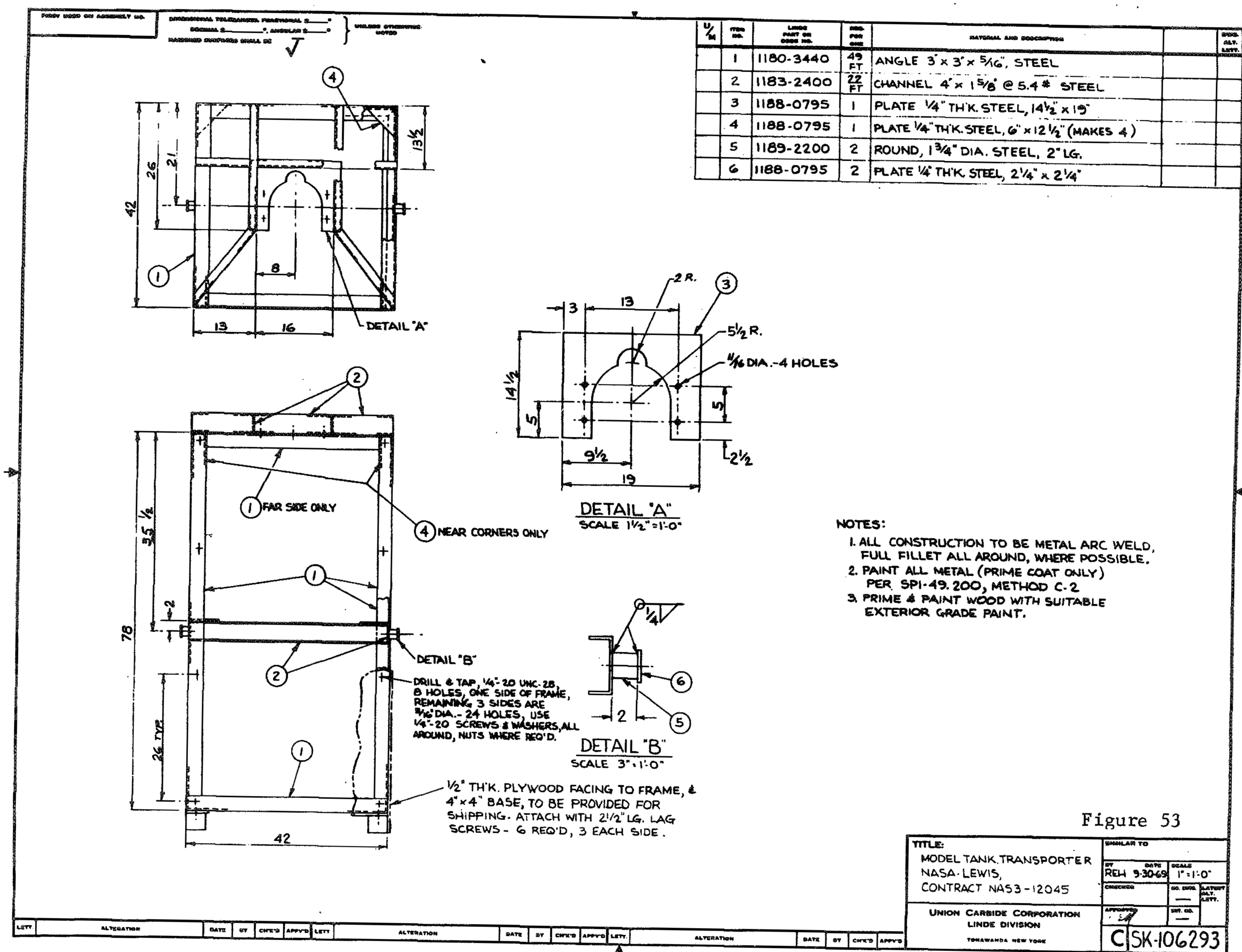


8. TECH. OBJECTIVE % COMP.

NASA-C-63 (Rev. 8-67)

NASA APPROVED SCHEDULE
CONTRACTOR'S WORKING SCHEDULE

(Prescribed in LHB 2300.1)



ITEM NO. | LINDE PART OR CODE NO. | REQ. PER ONE | MATERIAL AND DESCRIPTION
1 | 1180-3440 | 49 FT | ANGLE 3" x 3" x 5/16", STEEL
2 | 1183-2400 | 22 FT | CHANNEL 4" x 1 5/8" @ 5.4 # STEEL
3 | 1188-0795 | 1 | PLATE 1/4" TH'K. STEEL, 14 1/2" x 19"
4 | 1188-0795 | 1 | PLATE 1/4" TH'K. STEEL, 6" x 12 1/2" (MAKES 4)
5 | 1189-2200 | 2 | ROUND, 1 3/4" DIA. STEEL, 2" LG.
6 | 1188-0795 | 2 | PLATE 1/4" TH'K. STEEL, 2 1/4" x 2 1/4"
DETAIL "A"
2 R.
5 1/2 R.
11/16 DIA.-4 HOLES
DETAIL "A"
SCALE 1 1/2"=1'-0"
FAR SIDE ONLY
NEAR CORNERS ONLY
DETAIL "B"
DRILL & TAP, 1/4"-20 UNC-2B, 8 HOLES, ONE SIDE OF FRAME, REMAINING 3 SIDES ARE 5/16" DIA.- 24 HOLES, USE 1/4"-20 SCREWS & WASHERS, ALL AROUND, NUTS WHERE REQ'D.
DETAIL "B"
SCALE 3"=1'-0"
1/2" TH'K. PLYWOOD FACING TO FRAME, & 4"x4" BASE, TO BE PROVIDED FOR SHIPPING. ATTACH WITH 2 1/2" LG. LAG SCREWS - 6 REQ'D, 3 EACH SIDE.
26 TYP.
NOTES:
1. ALL CONSTRUCTION TO BE METAL ARC WELD, FULL FILLET ALL AROUND, WHERE POSSIBLE.
2. PAINT ALL METAL (PRIME COAT ONLY) PER SPI-49.200, METHOD C-2
3. PRIME & PAINT WOOD WITH SUITABLE EXTERIOR GRADE PAINT.
TITLE:
MODEL TANK TRANSPORTER
NASA-LEWIS,
CONTRACT NAS3-12045
UNION CARBIDE CORPORATION
LINDE DIVISION
TONAWANDA NEW YORK
REH 9-30-69
1"=1'-0"
C SK-106293

Figure 53

INST ENGR. ____ LERC ENGR. FADDOUL

FACILITY K RUN DATE ____ OPNS. ENGR. ____

ROCKET SYSTEMS DIVISION
RESEARCH INSTRUMENTATION REQUEST

NOTE 1: UPPER SQUARE "X" TO INDICATE RECORDING REQUESTED. LARGE SQUARE INDICATE SPECIFIC INSTRUMENT AND/OR CHANNEL NUMBER

PAGE 1 OF 3

ITEM NUMBER	PARAMETER	TRANS. MAKE OR TYPE	TRANS. MODEL	TRANS. SERIAL NUMBER	RECORDING RANGE	BAL. CHAN.	AMPLIFIER CH.	AMPLIFIER GAIN	SIGNAL CABLE LTR.	SIGNAL CABLE PAIR	D/A	DIG	FM	SC	SPD IN/MIN	OSCIL	SPD IN/SEC	FREQ / SPAN	BRUSH	SPD MM/SEC	CON-SOLE	MON PAIR	LIMIT %	REMARKS
1	SEMI PANEL TEMPERATURE	Cu/Const	T.C.		+150 °F -300 °F																			
2																								
3																								
4																								
5																								
6																								
7																								
8																								
9																								
10																								
11																								
12																								
13																								
14																								
15																								
16																								
17																								
18																								
19																								
20																								
21																								
22																								
23																								
24																								
25	UPPER HEAD + FLANGE TEMP																							
26																								

NASA P-1006 (10-66)

INST. ENGR. ____________ LERC ENGR. ____________

FACILITY ____ RUN DATE ________ OPNS. ENGR. ____________

ROCKET SYSTEMS DIVISION

RESEARCH INSTRUMENTATION REQUEST

NOTE 1: ☒ UPPER SQUARE "X" TO INDICATE RECORDING REQUESTED.
LARGE SQUARE INDICATE SPECIFIC INSTRUMENT AND/OR CHANNEL NUMBER

ITEM NUMBER	PARAMETER	TRANS. MAKE OR TYPE	TRANS. MODEL	TRANS. SERIAL NUMBER	RECORDING RANGE	BAL. CHAN.	AMPLIFIER CH.	AMPLIFIER GAIN	SIGNAL CABLE LTR.	SIGNAL CABLE PAIR	D/A	DIG	FM	SC	SPD IN/MIN	OSCIL	SPD IN/SEC	FREQ / SPAN	BRUSH	SPD MM/SEC	CON-SOLE	MON PAIR	LIMIT %	REMARKS
27	UPPER HEAD AND FLANGE TEMP	Cu/CONST	T.C.	~~+150°F~~ ~~-380°F~~	+150°F -380°F																			
28																								
29																								
30																								
31	TEST TANK TEMPERATURE																							
32																								
33																								
34																								
35		ROSEMONT			-300°F -423°F																			
36																								
37																								
38																								
39																								
40																								
41	SEMI PANEL PRESSURE	G.E.	22GT103		2.5 torr 10^{-10} torr																			
42																								
43	TEST LIQUID LEVEL (TOP)	UCC	2641-6		ON-OFF																			
44	TEST LIQUID LEVEL (MIDDLE)																							
45	TEST LIQUID LEVEL (BOTTOM)																							
46	TEST TANK PRESSURE				0-5 PSID																			
47	TEST TANK ATMOSPHERIC ΔP	ROSEMONT	1101D1-AECB		0-0.1 PSID																			
48	COLD GUARD LL (HIGH)	UCC	2641-6		ON-OFF																			ALL OF THE EQUIPMENT AND
49	COLD GUARD LL (LOW)																							INSTRUMENTATION FROM ITEM
50	COLD GUARD TEMPERATURE	ROSEMONT			-300°F -423°F																			#8 THRU 75 ~~##~~ ALSO TO
51																								BE USED UNDER R 3119, THUS
																								NO SPECIAL---

INST. ENGR. ______ LERC ENGR. ______

FACILITY ______ RUN DATE ______ OPNS. ENGR. ______

ROCKET SYSTEMS DIVISION

RESEARCH INSTRUMENTATION REQUEST

NOTE 1: [X] UPPER SQUARE "X" TO INDICATE RECORDING REQUESTED. LARGE SQUARE INDICATE SPECIFIC INSTRUMENT AND/OR CHANNEL NUMBER

ITEM NUMBER	PARAMETER	TRANS. MAKE OR TYPE	TRANS. MODEL	TRANS. SERIAL NUMBER	RECORDING RANGE	BAL. CHAN.	AMPLIFIER CH.	AMPLIFIER GAIN	SIGNAL CABLE LTR.	SIGNAL CABLE PAIR	D/A	DIG	FM	SC	SPD IN/MIN	OSCIL	SPD IN/SEC	FREQ / SPAN	BRUSH	SPD MM/SEC	CON-SOLE	MON PAIR	LIMIT %	REMARKS
52	SHROUD TEMPERATURE	ROSEMONT			+100 °F -110 °F																			(CONT.) REQUIREMENTS ARE TO BE MET EXCEPT
53				SEE REMARKS																				THAT THE SHROUD TEMPERATURES
54																								WILL REQUIRE A DIFFERENT
55																								BRIDGE SET UP.
56																								
57																								
58																								
59																								
60	FLOW METER TEMPERATURE	CU/CONST	TC		+180 °F -110 °F																			
61																								
62																								
63																								
64																								
65	WATER INLET TEMPERATURE	CU/CONST	TC		+150 °F 0 °F																			
66	COLD GUARD PRESSURE				0-5 PSID																			
67	COLD GUARD ATMOSPHERIC ΔP				0-0.1 PSID																			
68	FLOW METER	HASTINGS-RAYDIST			0-1 SCFH																			
69					0-10 SCFH																			
70					0-100 SCFH																			
71					0-0.1 SCFH																			
72					0-1000 SCFH																			
73	ATMOSPHERIC PRESS.	BAROMETER																						
74	CHAMBER PRESSURE																							
75																								
76	COLD GUARD PURGE LINE PRESSURE				0-30 PSID																			
77	COLD GUARD PURGE BASE CHAMBER ΔP	BARATRON			0-1 PSID																			

NASA P 1006 (10-60)

DIRS 02198

APPENDIX 9

TEST PROCEDURE
FOR THE EVALUATION OF THE
THERMAL DEGRADATION OF THE
SELF EVACUATING MULTILAYER INSULATION SYSTEM
DURING A SIMULATED LAUNCH ENVIRONMENT

Prepared by: Louis J. Demas 8/4/70

Louis J. Demas Date
Structural Dynamics Branch

Reviewed by: Harry P. Norris 8/28/70

H. P. Norris Date
Flight Program Office

Approved by: James R. Faddoul 9/11/70

J. R. Faddoul Date
LeRC Project Manager

Goddard Space Flight Center
Greenbelt, Maryland
July 28, 1970

NOTE

At the present time it is unlikely the LPS will be capable of vibrating the test item to the 8.5g level as depicted in this procedure. Since the maximum vibration level is not known and will not be known until the onboard checkouts of the vibration system have been completed, the 8.5g level was left in the procedure. Once the maximum vibration level is known the vibration test level for SEMI will be adjusted to that level if it is below 8.5g. There may also be certain frequencies where it will be impossible to vibrate due to LPS structural considerations. This will also be determined at the time of the onboard checkouts.

1. GENERAL

Test Title: Simulated Launch Environment Thermal Degradation Test on the Self Evacuating Multilayer Insulation (SEMI) System

Test Facility: Launch Phase Simulator (LPS) and associated instrumentation and support equipment.

Applicable Document: Self Evacuating Multilayer Insulation (SEMI) System Simulated Launch Environment Thermal Degradation Test Program Plan Report, J. R. Faddoul, NASA-Lewis Research Center

Job Order Number: 321-124-08-16-01

2. TEST OBJECTIVE

The purpose of this test is to determine if the launch environment will degrade the thermal performance of the SEMI system.

3. DESCRIPTION OF TEST ARTICLE

The test item will consist of the following:

(a) A 30 inch diameter by 60 inch long aluminum cryogenic test tank to which the insulation panels attach.

(b) SEMI panels consisting of alternate layers of foam spacers and double aluminized mylar radiation shields enclosed in a flexible vapor barrier. Each insulation panel is filled with gaseous CO_2. The panels are attached to the tank and each other with 3 inch by 3 inch nylon Velcro fasteners. The system is sealed on the outside with aluminum foil adhesive back tape. During the testing, the insulation panels will by cryopumped. This will be accomplished by filling the test tank with liquid nitrogen. This will cool down the tank walls and condense the CO_2 contained in the insulation panels, thus effectively evacuating the panels.

(c) A vibration adapter by which the tank attaches to the LPS vibration end cap. The adapter accommodates all necessary cryogenic plumbing for filling and venting of the LN_2.

4. TEST SETUP

After uncrating of the test item and prior to any testing, a complete inspection of the test item will be performed by the members of the SEMI crew. Any damage will be noted and still photographs of the damaged area will be taken.

The test will be conducted utilizing the acoustic, vacuum, steady-state acceleration and mechanical vibration capabilities of the LPS. The test item will be mated to the LPS vibration end cap and all instrumentation attached. Section 10 contains the tank handling procedure. Upon verification of all instrumentation, low level sine tests shall be conducted on the LPS in the off-board vibration mode. The purpose of these tests is to aid in finalizing the vibration specification. After completion of the tests, the end cap/test item combination will be rotated 90^{o} to the horizontal and attached to the LPS test chamber by means of the LPS loading vehicle.

5. TEST LEVELS

The desired test levels and profiles are shown in Figures 54 through 58. The specifications for the vacuum, acoustic and steady-state acceleration environments are within the capabilities of the LPS and should be met with little difficulty. The desired vibration specification is to maintain a constant 8.5g's from 10 to 100 Hz at the test tank c.g.; at the present time it is unlikely the LPS will be capable of vibrating the test item to this 8.5g level. Since the SEMI test will be the first test utilizing the on-board vibration capabilities of the LPS and the on-board checkouts have not been performed at the time of the writing of this procedure, there is limited knowledge as to the on-board capabilities. Off-board vibration system checkouts have indicated that the LPS vibration capabilities will be limited to a maximum level of 5g's and a frequency range of 5 to 60 Hz for both on-board and off-board testing. At the completion of the on-board checkouts these numbers may change and the SEMI vibration specification will be adjusted accordingly.

The following sections are written utilizing the desired levels and not the expected levels.

6. TEST DESCRIPTION - DESIRED VIBRATION LEVELS

This test will be performed in one phase only, subjecting the SEMI system to the combined launch environments of acoustic noise, mechanical vibration, steady-state acceleration and vacuum venting as depicted in Figures 54, 55 and 56. The specified octave band acoustic noise levels are shown in Figure 57 and sinusoidal vibration specification in Figure 58.

A four point vibration control shall be used for this test with accelerometers located at 0^{o}, 90^{o}, 180^{o} and 270^{o} on the shaker table at the base of the test item. See Figure 59 for location of accelerometers.

Prior to the combined test, a number of low level vibration tests will be performed in the off-board and on-board LPS vibration mode. Since it is desired to maintain a constant 8.5g's at the tank center of gravity and we can only control at the base of the test item, these low level runs will enable us to develop a vibration specification which will meet this requirement. These tests and the method for developing the specification will be as follows:

(a) A ½g sine survey in the X axis from 10 - 100 Hz will be performed in the off-board LPS vibration mode. The data recorded for the tank center of gravity and input monitor accelerometers will be analyzed. Transmissibility plots will be made with the tank center of gravity accelerometer as the denominator and the input monitors as the numerator. From these transmissibility plots, a vibration specification for maintaining a constant 4.25g's at the tank center of gravity will be developed by multiplying the transmissibilities by 4.25.

(b) A 4.25g vibration test in the LPS off-board mode using the specification developed above will be performed. The data from the tank center of gravity and input monitor accelerometer will be analyzed to insure that the 4.25g level was maintained at the tank center of gravity. If this level was not maintained, the necessary adjustments to the specification will be made and the test repeated.

(c) Once all off-board vibration tests have been completed, on-board vibration tests at ½g and 4.25g's will be performed with arm rotating at 5 RPM. Any final adjustments will be made. If these adjustments are significant, this test will be repeated. The specification will be changed to the 8.5g level for the combined environment test.

Upon completion of the low level tests, the SEMI tank will be filled with LN_2 in order to lower the tank temperature to between $-280^{o}F$ to $-300^{o}F$. While this is being done, a one-hour GN_2 (gaseous nitrogen) purge of the LPS test chamber will be performed. When the tank reaches the desired level, the LN_2 will be drained from the tank, the LN_2 lines disconnected and equipment removed from the rotunda. The test then will be performed as per Test Run Sequence (Section 9). After the test, the chamber will be returned to atmospheric pressure over a half-hour interval at a rate of approximately 24 torr/minute by backfilling with GN_2. The test item will be removed from the LPS chamber and a thorough post-test inspection shall be performed by members of the SEMI crew. Still photographs will record all anomalies. The LPS chamber will be inspected for any evidence of debris or insulation particles immediately after the test.

7. INSTRUMENTATION

The instrumentation planned for the test is as follows:

Temperature

Twelve temperature measuring devices will be recorded during the test. Six will be Rosemont temperature sensors mounted on the tank wall and six will be copper-constantan thermocouples, three each on two insulation panels. These devices will be provided and installed by Lewis Research Center and GSFC will be responsible for recording only.

Vibration

A total of fifteen accelerometers will be used during the test. Four accelerometers (Endevco type 2224) will be located on the shaker table at the base of the test item to monitor the four control accelerometers. Two accelerometers (Endevco type 2221D) will be mounted adjacent to accelerometer number 1 oriented to measure the lateral crosstalk vibration in the X and Z axes. A single accelerometer (Endevco type 2221D) will be mounted on the shaker table as a dump monitor. Three accelerometers (Endevco type 2271A) will be mounted at the tank/vibration adapter flange, measuring the response in the X, Y and Z axes. Three accelerometers (Endevco type 2222B) will be attached at the tank center of gravity measuring the response in the X, Y and Z axes. Two accelerometers will be supplied by Lewis Research Center and will be located underneath the test insulation on the dummy support structure. Figure 6 shows accelerometer location and number.

Acoustics

Five B&K (type 4136 or equivalent) microphones are to be located in the LPS chamber as shown in Figure 60. Microphone number 3 will be the control microphone.

Pressure

One CEC type 4-312, 0 - 15 psia pressure pickup will be used to monitor the pressure profile inside the LPS chamber. One 0 - 100 psi differential pressure transducer will be installed on the Vibration Test Flange Assembly to read internal tank pressure.

Visual

Two closed circuit television cameras will be located in the chamber for real time observation of the test. Each camera will be oriented normal to the Y-Z plane and one will look along the +Z edge of insulation, the other, the -Z edge of insulation. The output of both cameras will be displayed in the control room and one camera's output will be recorded on video tape. One 16 mm movie camera (film speed 64 frames/second) shall be used during the test. Timing marks at 0.10 second intervals shall be provided on the film.

8. DATA ACQUISITION

Data from all transducers will be recorded on magnetic tape during the test run. In addition the six Rosemont temperature transducers, both the chamber and tank pressure, and accelerometers 1X, 2X, 6X, 6Y, 6Z, 7X, 7Y and 7Z will be simultaneously presented on oscillograph. The control microphone will be analyzed by octave bands and indicated on oscilloscope.

All records will be fully labeled (as will the tape be annotated) with the following information:

SEMI System - LPS Test

Date and Run Number

Transducer and Number (i.e., accelerometer # 1X)

Channel Allocated

Exposed motion picture film from the test will be promptly processed and examined. Three copies of the film will be made and all film containers and reels will be appropriately identified as indicated above.

9. DATA ANALYSIS

Temperature

Temperature versus time plots for each temperature transducer are required.

Vibration

Filtered and unfiltered plots (g level versus frequency) of each accelerometer response are required.

Acoustics

All microphone responses shall be presented as time histories, i.e., Overall Sound Pressure Level (OASPL) versus time. In addition, an Octave Band Analysis of the response of each microphone during each stationary OASPL interval is required. Narrower filter bandwidth analysis may be specified for certain locations after inspection of broad filter band data.

Pressure

Pressure versus time plots are required for both pressure transducers. These will be presented as X-Y plots with abscissa being time in seconds and the ordinate being pressure in torr.

10. TEST RUN SEQUENCE

The following sequence will be followed for the SEMI testing:

1. Uncrate and visually inspect the test item prior to installation on the vibration end cap.

2. Install microphones, pressure transducer, movie camera and television cameras in LPS chamber.

3. Attach test item to vibration end cap, as per handling procedure (Section 11).

4. Install and check out all instrumentation (except for three accelerometers on tank center of gravity).

5. Connect LN_2 fill and drain and vent lines.

6. Cool inside of tank with LN_2 and attach three accelerometers at tank center of gravity.

7. Perform low level sinusoidal survey.

8. Reduce survey data and prepare 4.25g vibration specification.

9. If required cool inside of tank with LN_2 in order to maintain vacuum in panels.

10. Perform 4.25g vibration test in off-board mode.

11. Reduce data and make any necessary adjustments. Repeat 4.25 test if necessary.

12. Attach end cap/test item combination to LPS test chamber.

13. Connect external LN_2 plumbing.

14. Start LN_2 fill.

15. Monitor Rosemont output until all sensors read between $-280^{o}F$ to $-300^{o}F$.

16. Drain LN_2 from tank.

17. Disconnect plumbing and remove equipment from rotunda and close doors.

18. Rotate arm at 5 RPM and perform ½g survey and 4.25g vibration test.

19. Reduce data and make adjustments to specification.

20. Repeat test if necessary.

21. Repeat steps 13 through 17.

22. Start recorders and initiate test programs.

23. At completion of test, stop recorders.

24. Start backfill of chamber with GN_2 at 24 torr/minute rate.

25. Monitor Rosemont sensors until tank warmup is complete.

26. Remove end cap/test item combination from the LPS test chamber.

27. Visually inspect the insulation system for any anomalies.

28. Inspect LPS test chamber for debris and bits of insulation.

29. Perform post-test instrumentation calibration.

30. Disconnect plumbing and instrumentation.

31. Remove tank from end cap and recrate for shipment to Lewis Research Center.

11. HANDLING PROCEDURE

The following is the SEMI test tank handling and attachment procedure. The actual handling of the tank will be done by members of the SEMI crew with GSFC personnel operating the crane and providing necessary support.

1. Attach vibration test adapter to vibration end cap.

2. Attach vibration test flange assembly to test vessel flange.

3. Attach lifting rig to trunnion pins provided for on the shipping containers for lifting and handling.

4. Lift tank and shipping containers and invert them.

5. Position tank directly over vibration test adapter and lower into position.

6. Attach tank to vibration test adapter with four of the eight required bolts.

7. Remove bolts which hold shipping containers to tank.

8. Remove shipping containers and store until test is completed.

9. Install the other four required bolts.

When removing the tank from vibration end cap the above procedure is reversed.

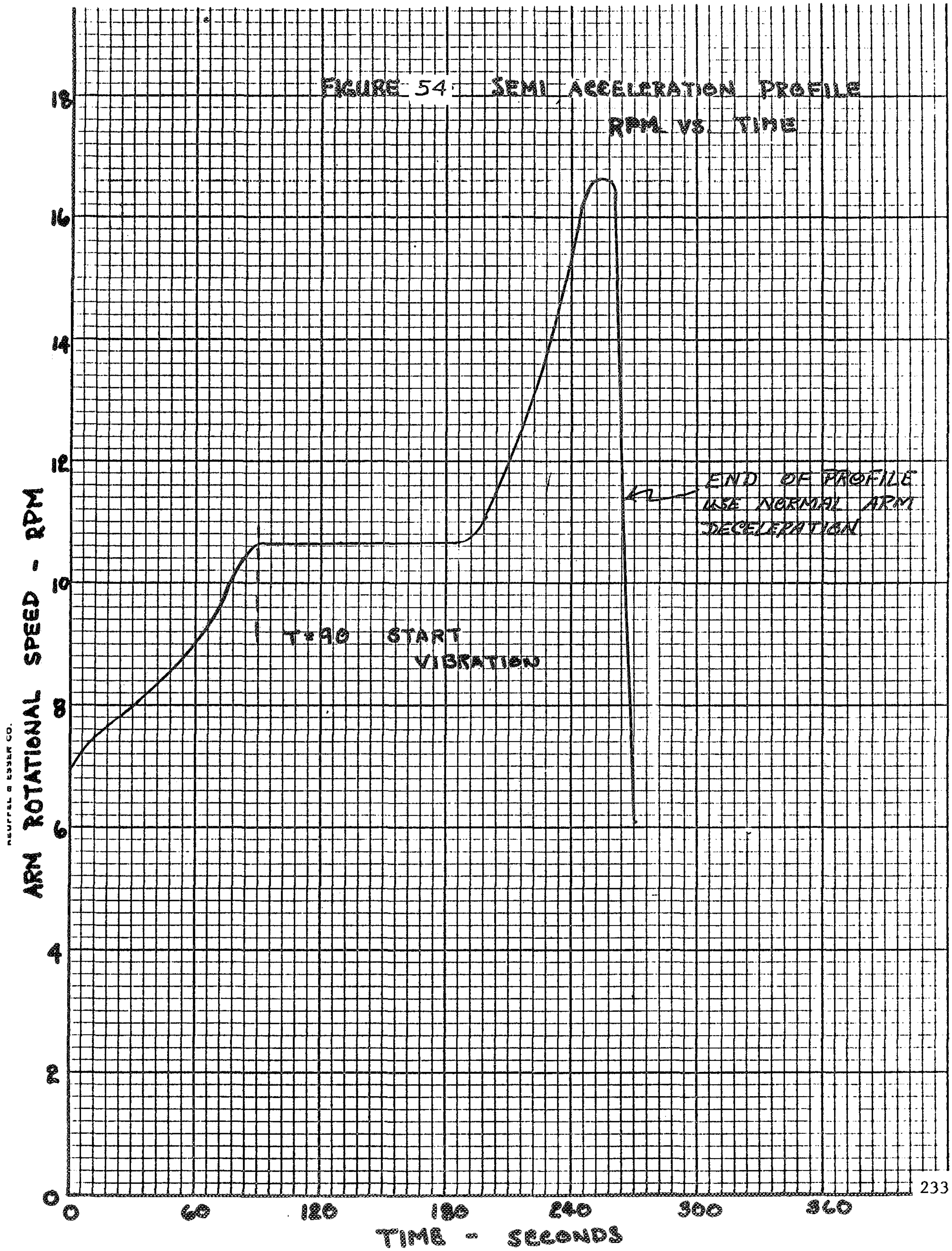


FIGURE 54 SEMI ACCELERATION PROFILE
RPM VS. TIME
END OF PROFILE
USE NORMAL ARM
DECELERATION
T=90 START
VIBRATION
ARM ROTATIONAL SPEED - RPM
TIME - SECONDS
0
2
4
6
8
10
12
14
16
18
0
60
120
180
240
300
360
KEUFFEL & ESSER CO.

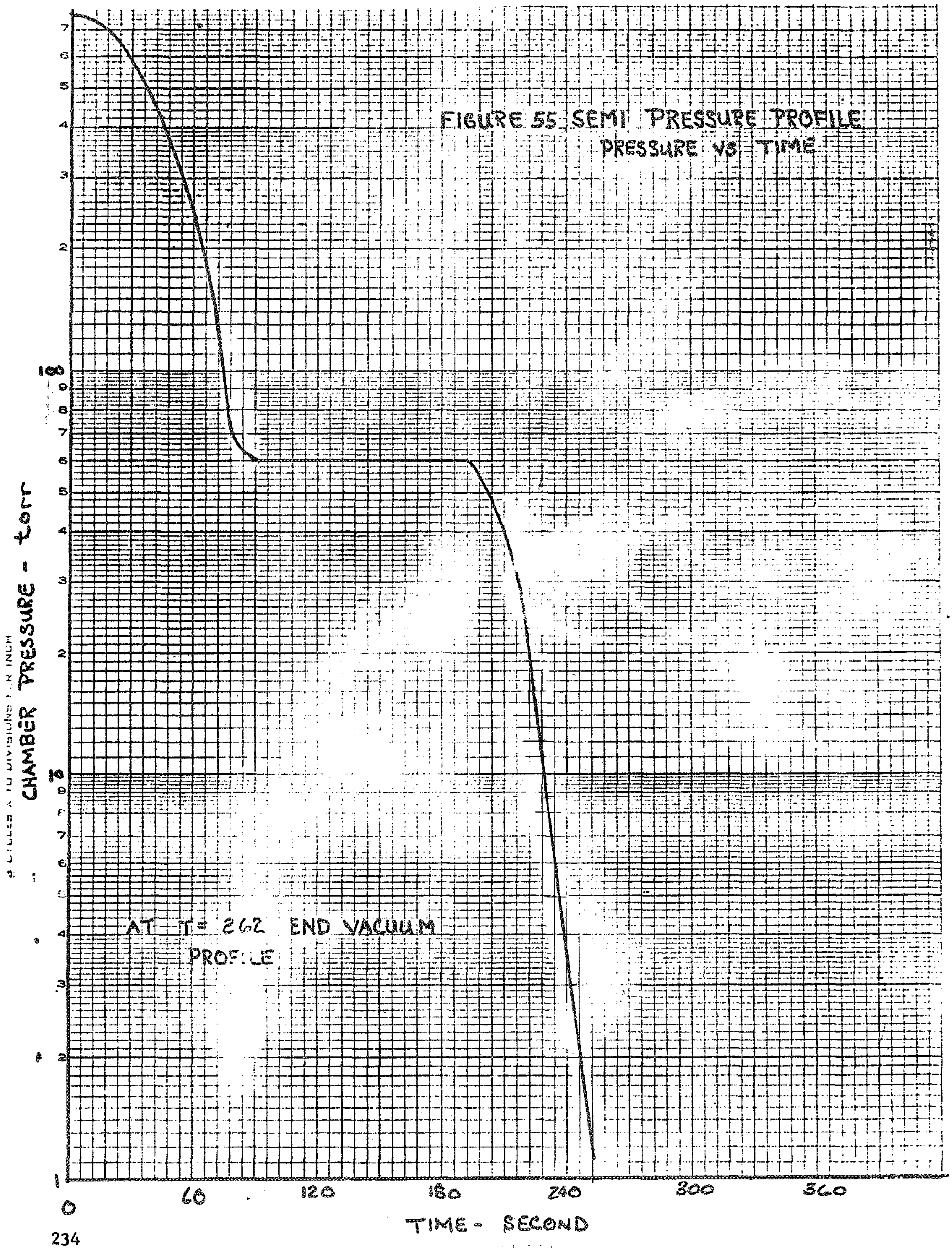


FIGURE 55 SEMI PRESSURE PROFILE
PRESSURE VS TIME
CHAMBER PRESSURE - torr
AT T= 262 END VACUUM
PROFILE
TIME - SECOND
0
60
120
180
240
300
360

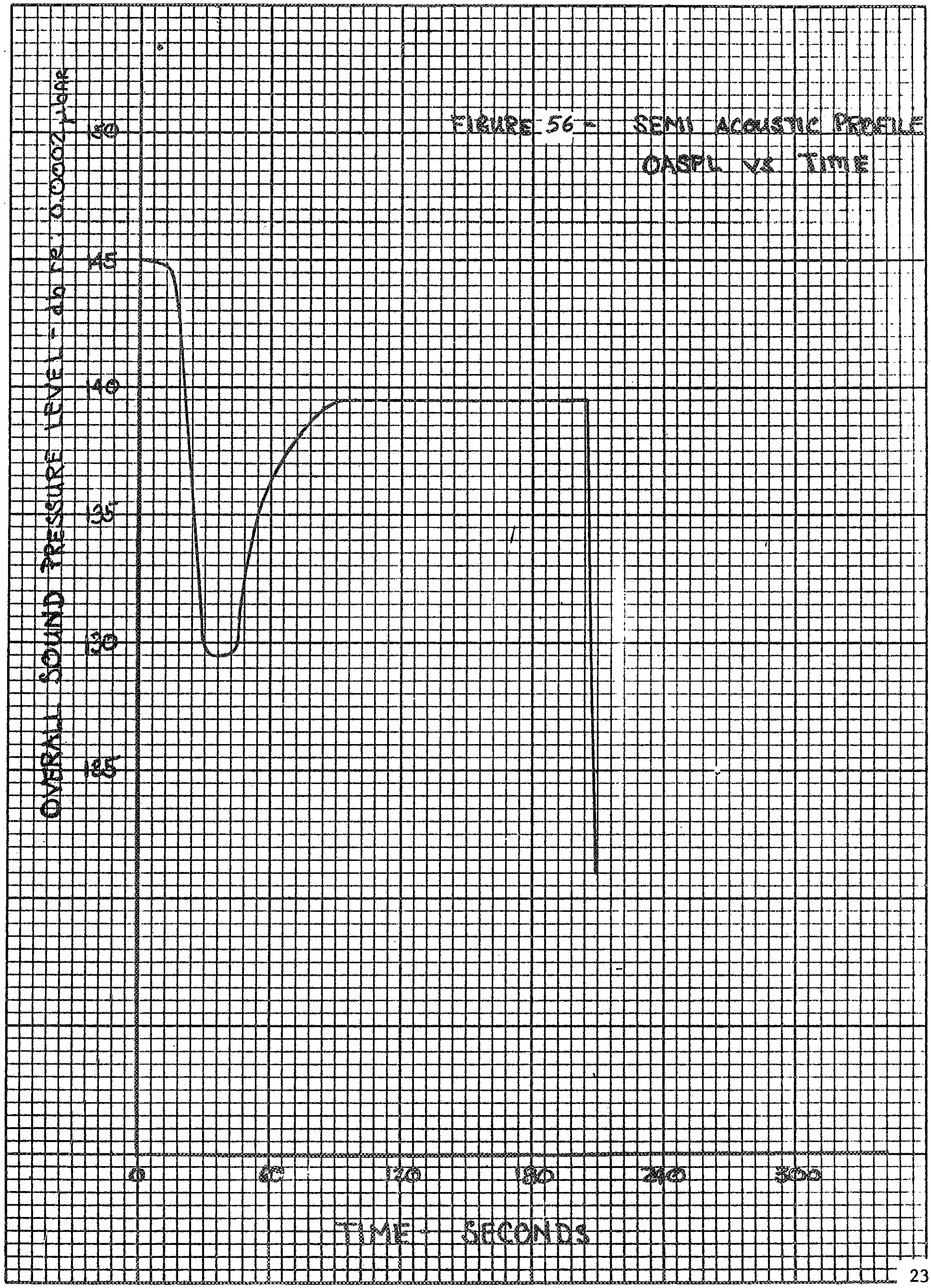


FIGURE 56 - SEMI ACOUSTIC PROFILE
OASPL VS TIME
OVERALL SOUND PRESSURE LEVEL - db re: 0.0002 μbar
150
145
140
135
130
125
0
60
120
180
240
300
TIME - SECONDS

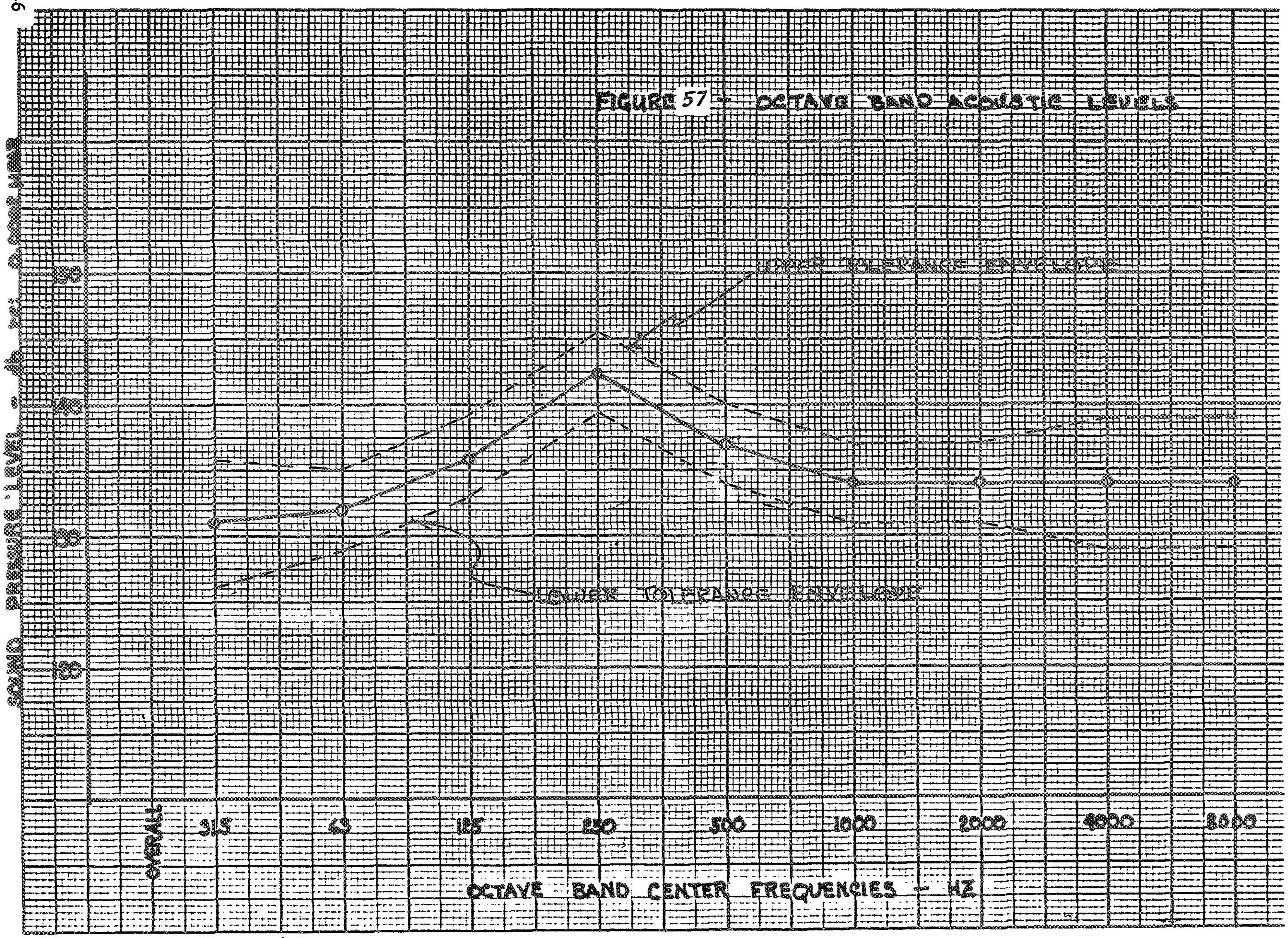
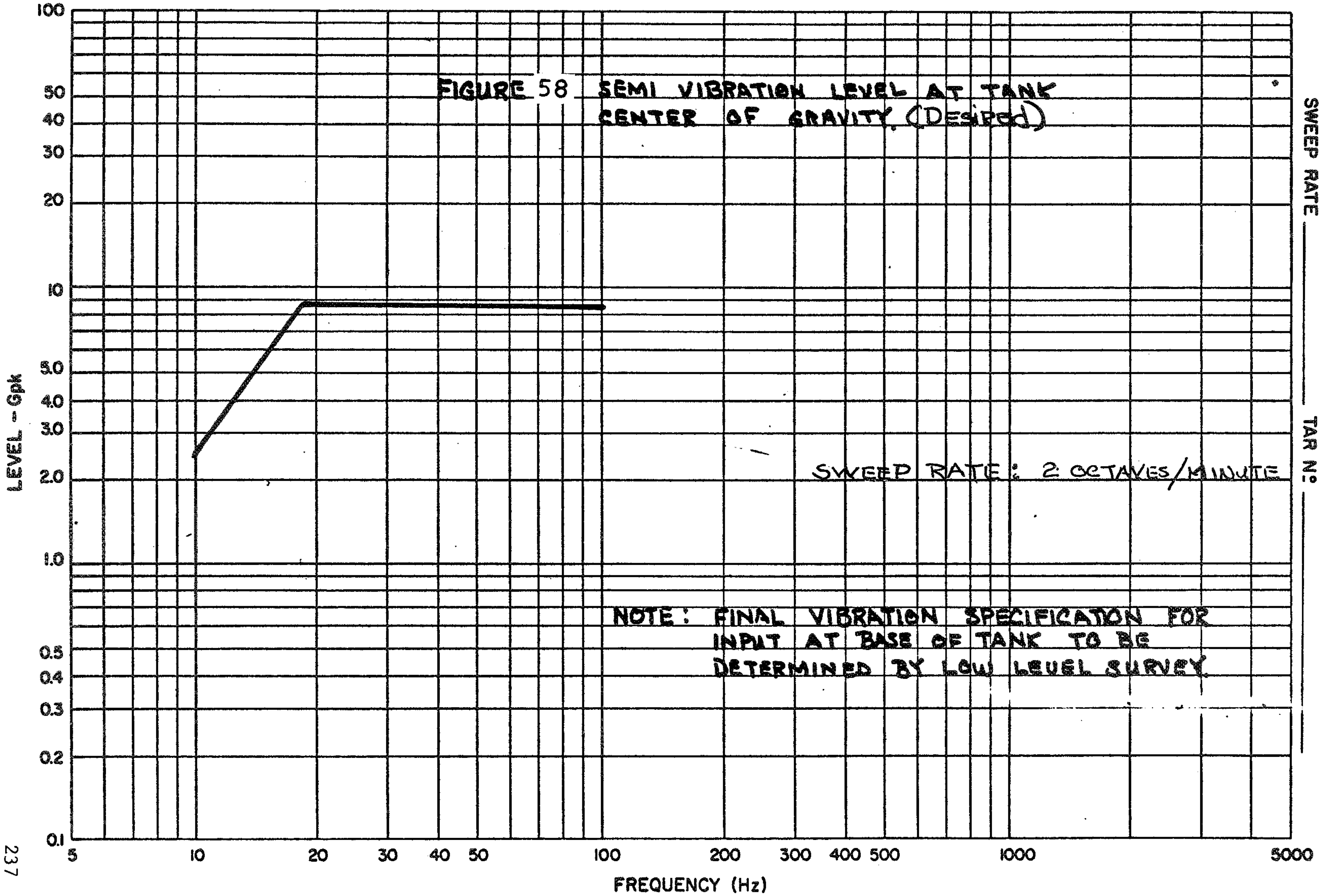


FIGURE 57 - OCTAVE BAND ACOUSTIC LEVELS

STRUCTURAL DYNAMICS VIBRATION LABORATORY - GSFC

PROJECT SEMI AXIS X OPERATOR ——

ITEM TEST TANK DATE ——



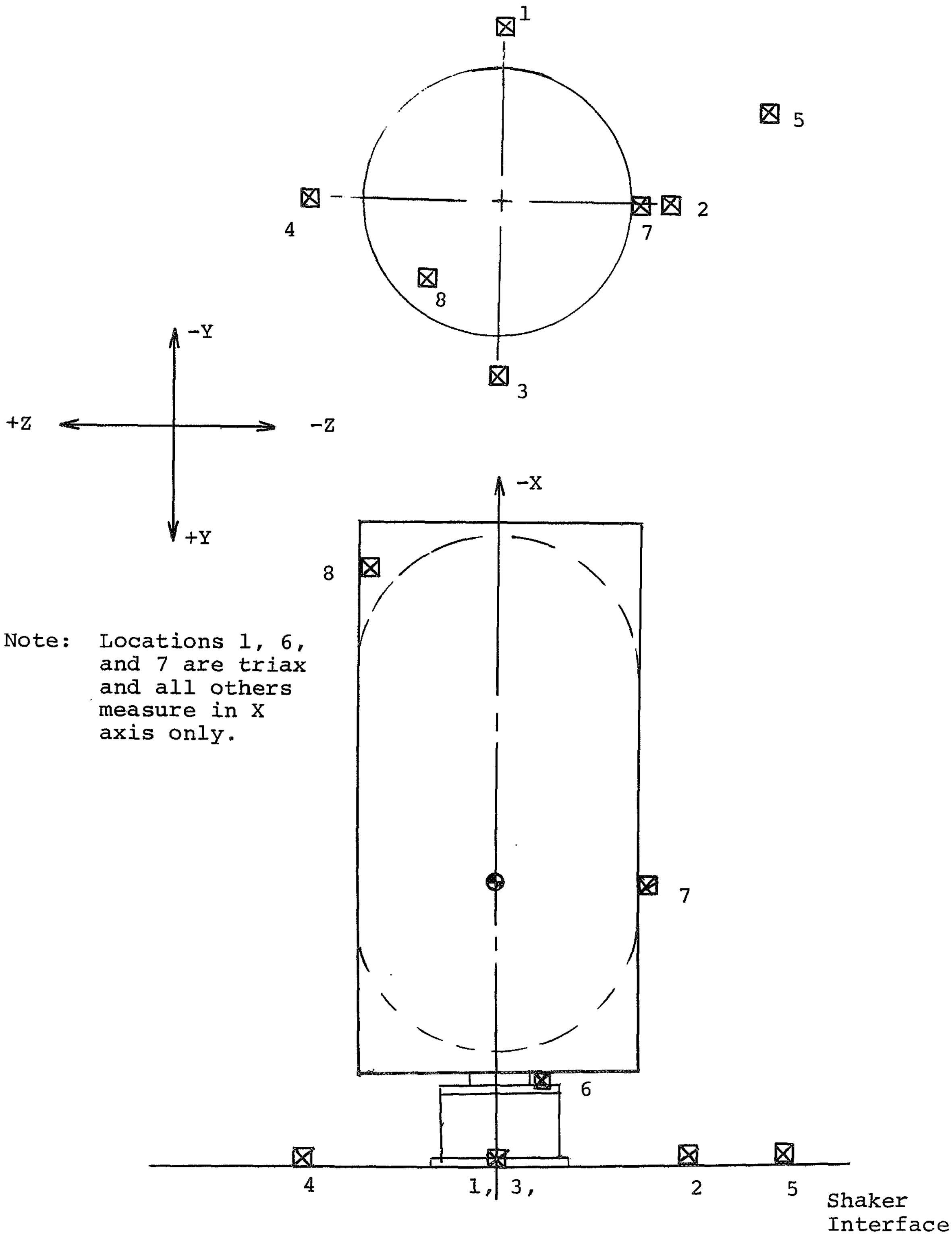


Figure 59 Test Configuration and Accelerometer Location

Mike No.	1	2	3	4	5
Station	77	29	29	29	29
Angle	℄	0	90	180	270
Projection*	69	27	27	27	27

* distance from wall to mike

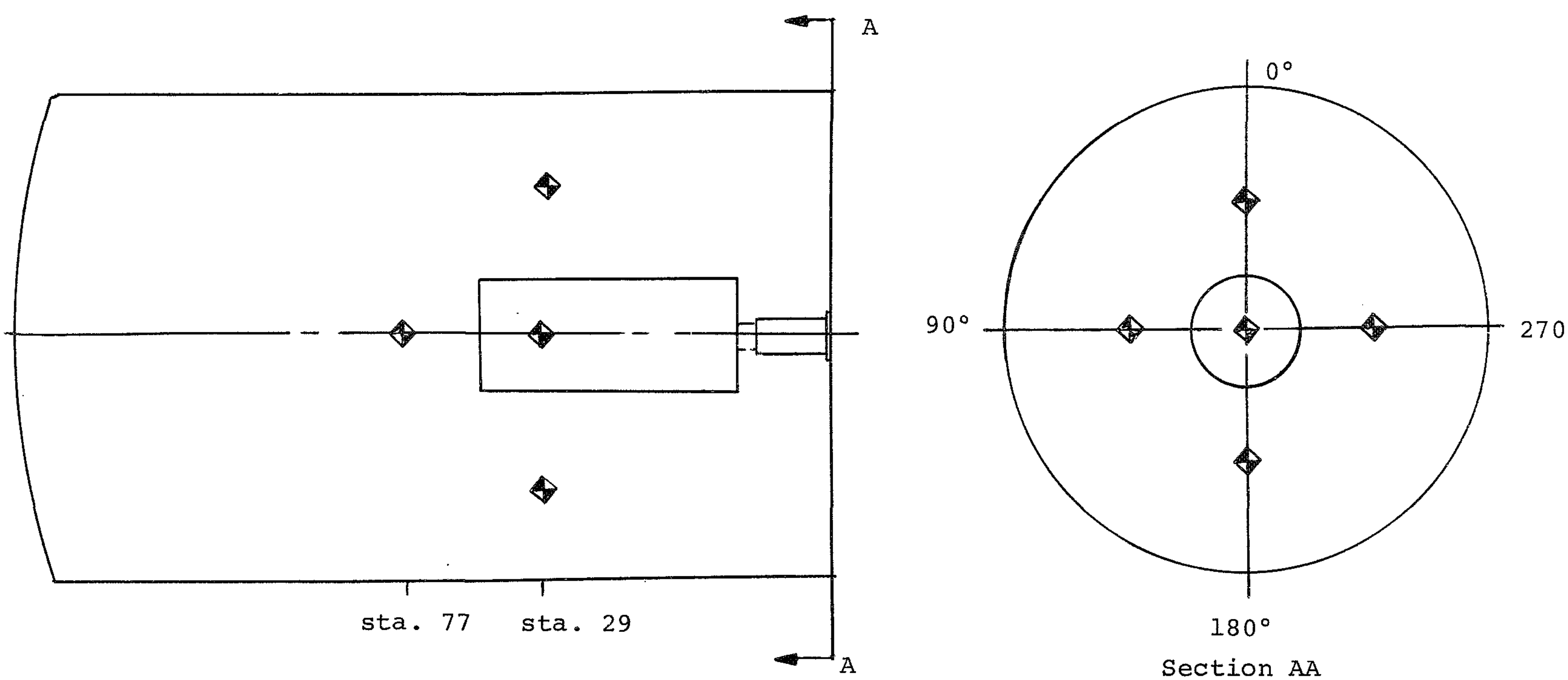


Figure 60 LPS Microphone Location

APPENDIX 10

TEST PLAN - SUBSCALE VIBRATION TESTING, TASK I

In order to verify the validity of the structural analysis, Linde plans to perform vibration testing of several subscale test panels, as provided for in the Task I work statement. The tests are described below.

OBJECTIVE

The objective of these subscale tests are to evaluate the effect of vibration on panels using punched hole polyurethane foam spacers and the Velcro fasteners that are used to attach the panels to each other and to the test fixture.

TEST PANEL DESIGN

One of the three test setups is shown in Figure 61. The second configuration to be tested will be the same except for the direction of shingle. The vibration fixture is of a design that allows the 1 ft. x 2 ft. panel to be mounted in a plane having the 2 ft. axis vertical as shown or placing the 2 ft. axis horizontal. This will permit testing a particular panel for both directions of shingle without removing the panels from the metal plate between tests. By rotating the vibration machine 90°, it will be possible to test the panel for conditions simulating the insulation applied to the bottom of the tank at the Goddard tests, i.e. forced vibration applied parallel to the layers in combination with a one "g" force applied downward and normal to the insulation.

Two dummy panels will be used to simulate panel shingling. A vacuum common to both of the dummy panels and the test panel will be attained via an evacuation port from the back side of the metal panel. The outer (test) panel will be contact cemented directly to the front of the metal plate to provide a vacuum seal. The exposed outside test panel casing will be 0.001 inch clear Mylar rather than the 4-ply aluminized Mylar laminate. This permits visual observation of the foam spacers and Mylar radiation shields during the test. The test panel will contain six composite foam spacers and five Mylar radiation shields. The dummy panels will contain five solid layers of 0.1 inch thick open cell rigid foam separated by aluminized Mylar radiation shields.

TEST SYSTEM COMPONENTS

Spacer Configuration

Each test panel will utilize the PT-6 punched hole configuration (Contract NAS 3-7953), i.e. a three layer composite spacer consisting of two .02 inch thick open cell rigid polyurethane foam layers containing punched holes and one .02 inch thick layer of unpunched open cell rigid polyurethane foam. The two punched hole layers are positioned relative to each other such that support is achieved only at the intersection of the two webs. The PT-6 configuration has 1-1/4 in. square holes located on 2 in. centers.

Panel Seal

The panels will be fabricated of a 4-ply laminate of aluminized Mylar casing material and bonded together with contact cement. This procedure will provide sufficient vacuum integrity for ambient temperature vibration testing. This will also eliminate the necessity of making Narmco adhesive joints, which are more complicated to fabricate, yet would not contribute to these tests at ambient temperature.

PANEL-TO-PANEL AND PANEL-TO-TANK ATTACHMENT

Based on the results of the structural analysis, Velcro fasteners are recommended for panel-to-panel and panel-to-tank attachments to compensate for deflections caused by thermal and evacuation considerations. The relative motion between panels can be easily contained within the Velcro fastener and will not place the panel casing in tension to the extent as would be experienced if the SEMI panels were bonded intimately together with adhesive.

INSTRUMENTATION

Testing will be performed on a Unholtz-Dicke 1200 lb Model 56 vibration machine. A Kistler accelerometer Model 808A, Serial No. 538 mounted on the plate as well as the machine mounted accelerometer Endevco type 2224C, Serial No. JG-87 will be used to measure vibration loadings.

A 0 to 800 mm Hg Wallace and Tiernan Model FA160, Serial No. FF 02100 absolute pressure gauge will be used to indicate panel vacuum during testing.

TEST CONDITIONS

All vibration testing will be performed at ambient conditions. The panels will be tested in three directions for either or both evacuated and recovered insulation conditions. The sweep time, frequency "g" level, and range will be as follows for all tests, as listed below.

EXPLORATORY

One "g" sweep from 20 to 150 Hz in 60 seconds

Time (Seconds	Frequency
0	20
20	40
30	50
40	90
50	110
60	150

TEST

8.5 "g" sweep from 20 to 150 Hz in 60 seconds

Repeat above sweep

Test No. 1 Panel Evacuated - Shingle vertical

Test No. 2 Panel Recovered - Shingle vertical

Test No. 3 Panel Recovered - Shingle horizontal

Test No. 4 Panel Recovered - Shingle horizontal - traverse to vibration - 1 "g" loading acting normal to multilayer insulation

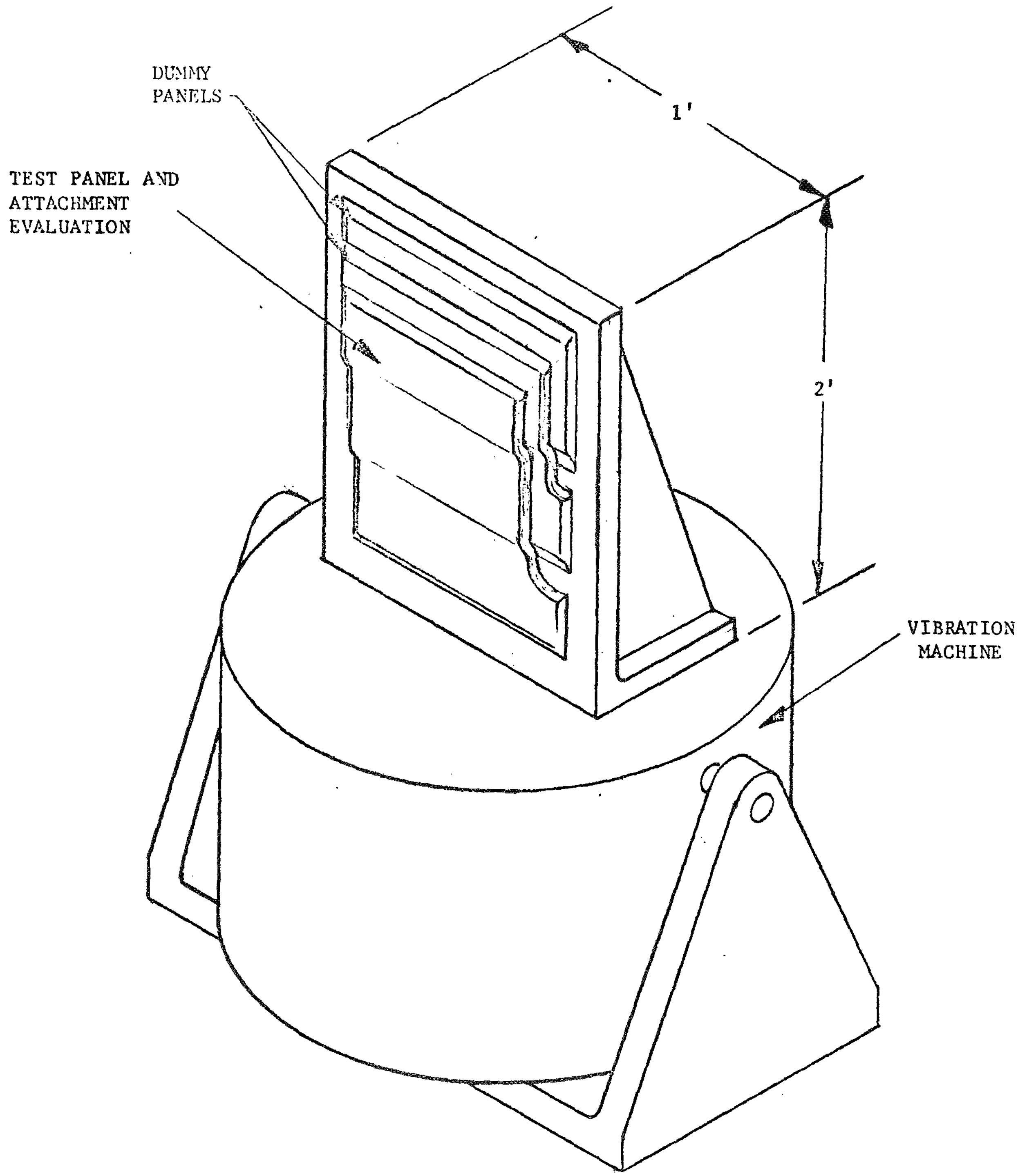


FIGURE 61 SCALE MODEL DYNAMIC TESTING

APPENDIX 11

VELCRO Fastener Qualification Tests

Results of the structural analysis of Task I indicated that a rigid adhesive bonding system between panels and tank or panels to panels would cause large stresses in the panel casing material, and would likely lead to panel failure. Instead of the rigid system, the analysis indicated that VELCRO fasteners would be acceptable since the fastener would permit some slipping between panels, to allow for thermal and evacuation stresses, while still maintaining panel placement. It, thus, became necessary to qualify an adhesive system for bonding the nylon VELCRO to aluminum plates and Mylar in addition to determining allowable stress levels for the VELCRO loop and pile closure assembled at 15 psi. (The 15 psi assembly pressure was chosen to simulate assembly of the panels equivalent to a one atmosphere ground hold condition.) Samples of VELCRO tested in tension were assembled at a 50 psi in compressive load to determine effect on the tensile strength achieved. Two adhesive systems were tested, namely vendor supplied VELCRO adhesive backed (VELCRO No. SA-0145A) and plain uncoated VELCRO using a urethane adhesive Narmco 7343/7139.

I. ADHESIVE JOINTS - PROCEDURE

A. Adhesive Backed VELCRO (VELCRO No. SA-0145A Coating)

VELCRO adhesive backing was wiped with liberal amounts of MEK, allowed to become tacky (approximately three minutes) and then placed on the clean substrate and allowed to cure overnight at ambient temperature and pressure.

1. Carbon Steel Substrate

   Roughened with emery cloth and wiped with MEK.

2. Aluminum Substrate

   Roughened on a wire wheel and wiped with MEK.

3. Mylar Substrate

   Wiped with MEK.

B. Narmco Bonded Joints

VELCRO

Narmco adhesive was applied to the 2 inch wide plain backed strips by brushing, following a wipe with MEK.

The Narmco was mixed to the following proportions.

7343	100 gms.
7139	11 gms.

When adhered to the aluminum and mylar substrates, the samples were cured at room temperature under 3 psi for 24 hours. Final cure was obtained at 0 psi and 170° for 4 hours.

1. Aluminum Substrate

The aluminum was prepared for priming by roughening with a wire wheel, and wiping with MEK.

A thin coat of Goodyear prime was applied by brushing after mixing to the following proportions:

Goodyear 207B	100 gms.
Toluene	63 gms.
MEK	27 gms.
Goodyear 207C	4 gms.

2. Mylar Substrate

The Mylar was prepared for Goodyear prime by wiping with MEK. A thin coat was applied by brushing. The primed aluminum and Mylar substrates were cured at room temperature for 24 hours.

II. CLOSURE TESTS

The 2 inch x 2 inch samples of VELCRO were joint under 15 psi closure pressure for all tests, with the exception of one run in tension using 50 psi closure pressure. Testing was performed using a calibrated 0 to 50 lb. spring scale to determine statically applied loads. Testing included 90° peel, shear stress and tensile stress. (See Figure 62) Samples were submerged in liquid nitrogen for 6 cycles before testing.

III. FASTENER TESTS

Adhesive joint tests were performed on both the VELCRO adhesive and the Narmco system. Tests consisted of shear and peel, (See Figure 63) using a 0 to 50 lb. calibrated spring scale.

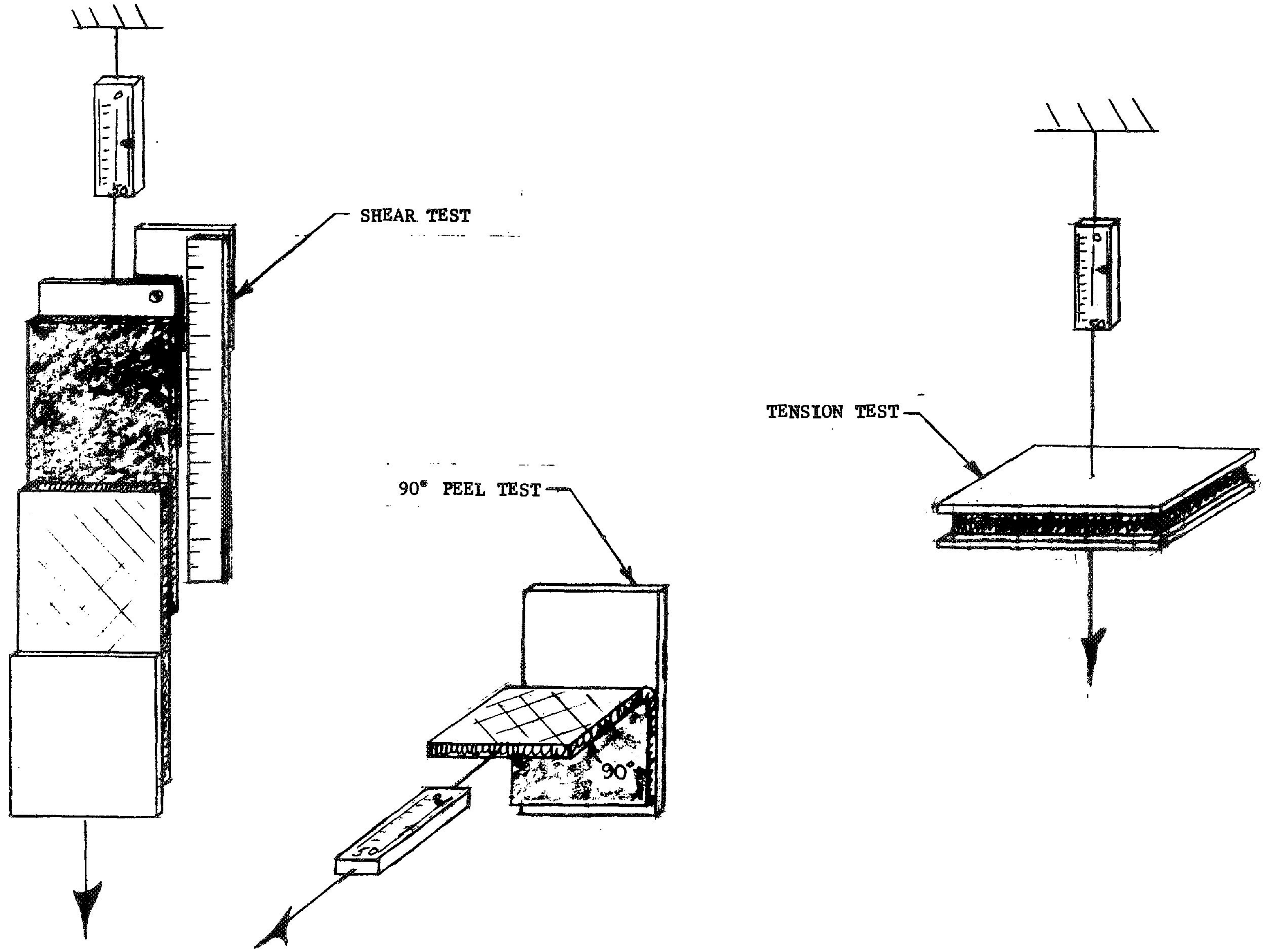
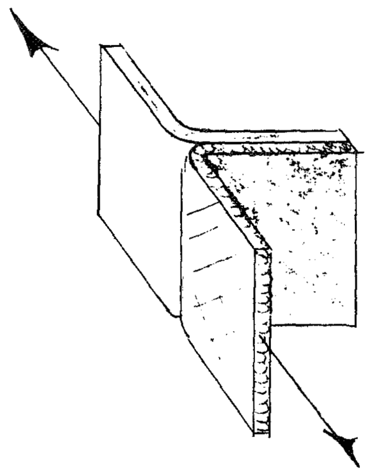
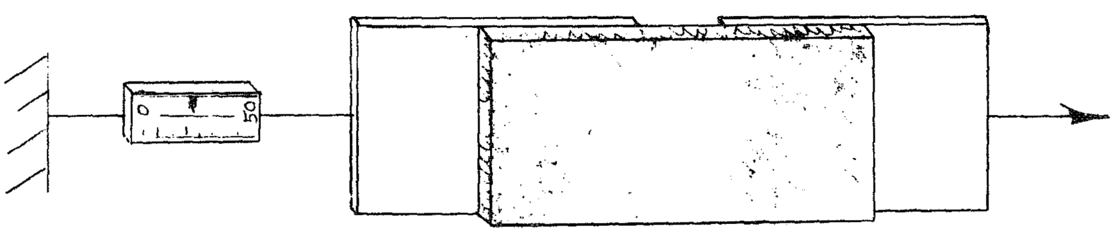


FIGURE 62

VELCRO LOOP AND PILE CLOSURE
ASSEMBLY TESTS



VELCRO TO MYLAR
180° PEEL TEST



VELCRO TO ALUMINUM SHEAR TEST

FIGURE 63

ADHESIVE TESTS

APPENDIX 12

CASING MATERIAL PERMEABILITY TESTS

Casing Material Permeability Tests

Helium permeability tests were performed with the use of the permeability tester shown in Figure 64.combined with a Vecco MS-9 helium mass spectrometer leak detector. The casing material to be tested is cut to a 6 inch diameter disk. This disk is placed between a double set of "O" rings in the permeability tester. The lower "O" ring provides a vacuum seal between the sample and the tester. A porous bronze disk provides support for the casing material sample when one side of the sample is evacuated with the leak detector.

The procedure of conducting a permeability test is as follows: The casing material sample is cut to the required diameter and placed between the "O" rings which are compressed from a seal by tightening the wing nuts. The permeability tester is then connected to the leak detector and a vacuum is pumped on one side of the sample. When the sample has been pumped to approximately 1×10^{-5} torr, the leak detector scale is zeroed and a standard leak, mounted on the side of the leak detector, is opened into the system, causing the leak detector scale to indicate the number of units proportional to the standard leak. This value is recorded for use in the calculation to determine the sample permeability. The standard leak is then valved off from the system, and it is noted that the detector scale returns to zero. To assure that the lower vacuum "O" ring seal of the permeability tester is not leaking, a spray of helium is put around the outside of this "O" ring. Any leakage would be immediately indicated on the leak detector scale. After assuring that leakage is not present, helium is purged across the top of the casing material sample through the tubing in the top of the permeability tester. When the leak detector scale has reached a steady-state value, this value is recorded and the test is thus completed. To calculate the permeability of the test sample, the following equation is used:

$$\text{Permeability of sample } \frac{\text{Atm cm}^3}{\text{sec-ft}^2}$$

$$= \frac{\text{Steady-state detector scale reading (units)}}{\text{Standard leak detector scale reading (units)}} \times \frac{\text{Value of standard leak (Atm cm}^3\text{/sec)}}{\text{Area of test sample (ft}^2\text{)}}$$

Note: The above equation assumes use of 100% helium test gas. The actual permeability is inversely proportional to the helium concentration in the test gas.

Example: For a 1% helium test mixture, the actual permeability is 100 times the measured permeability.

FIGURE 64 Permeability Tester Disassembled (1286-65)

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